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Survey on Practical Technologies and Applications for High Temperature Heat Pumps Intended to Recycle Energy

Thesis submitted for examination for the degree of Master of Science in Technology.

Espoo 24.04.2020

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Abstract of the master's thesis

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Title of thesis Survey on Practical Technologies and Applications for High Temperature Heat Pumps Intended to Recycle Energy

Master programme Advanced Energy Solutions **Code** ENG3068

Thesis supervisor Prof. Ville Vuorinen

Thesis advisors MSc Ari Aula & MSc Aleksi Aarnio

Date 24.04.2020 **Number of pages** 72+9 **Language** English

Abstract

The objective of this thesis was to find out practical high temperature heat pump (HTHP) technologies and potential applications for heat sinks at least up to 100 °C within a heating capacity range of 50...1000 kW along with to find out the profitability of HTHPs. A literature survey was executed to find out the technologies and applications. The survey was focusing on both the state-of-the-art research and practices. The profitability was found out via a Finnish case study. According to the survey, there are at least 32 commercialized HTHP models currently from which several can produce heat sinks of at least 100 °C. Several industrial sectors have some processes whose heat demand could be supplied with HTHPs that can produce heat sinks up to 100 °C. In addition, current district heating networks are a promising application for HTHPs since the heat demand is generally at temperatures below 100 °C. According to the survey, the refrigerant, the compressor and the cycle configuration are the most critical considerations as for HTHPs. There is both scientific and practical evidence on that the HTHP technology exists for heat sinks of at least 100 °C, even up to around 165 °C. The refrigerant should be selected such that its critical temperature is at least 10...15 °C higher than that of the condensation temperature and the vapor pressures should be such that the compressor can withstand those, both at the suction and at the discharge. In principle, the refrigerant selection determines the higher temperature limit that could be supplied with a HTHP, at least up to 165 °C. If considering higher temperatures, the maximum compressor discharge temperature of around 180 °C seems to be a limiting factor that is a technological challenge. In outline, the temperature difference between the condensation and evaporation temperatures (temperature lift) determine whether to use a single-stage or a two-stage compression. As for most of suitable refrigerants, the use of an internal heat exchanger (IHX) is suggested to provide the minimum superheat and to increase both the coefficient of performance and the volumetric heating capacity at least up to 20 % along with this improvement is the same in the operational emissions and the energy efficiency. In general, temperature lifts up to 60 °C can be feasible with the single-stage compression. Higher temperature lifts, up to even 120 °C, should be implemented with the two-stage compression to avoid a decrease in the performance. In addition, the two-stage compression can be used to limit the discharge temperature that is especially important as for some refrigerants along with when using the IHX. According to the conditions of the case study, HTHPs can be currently on average around 4.5 times more environmentally friendly and around 1.5...5.5 times more energy efficient when compared to the Finnish average district heating. As for the case study, the HTHPs are economically profitable as well because the simple payback period was estimated to be around two years. However, the profitability is highly sensitive with respect to the source information.

Keywords refrigeration technology, high temperature heat pump, energy recycling, refrigerant, COP

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Diplomityön tiivistelmä

Tekijä Timo Sarin

Työn nimi Selvitys käytännönläheisistä kuumalämpöpumpputekniikoista ja käyttökohteista tarkoituksena kierrättää energiaa

Maisteriohjelma Advanced Energy Solutions **Koodi** ENG3068

Työn valvoja Prof. Ville Vuorinen

Työn ohjaajat DI Ari Aula & DI Aleksi Aarnio

Päivämäärä 24.04.2020 **Sivumäärä** 72+9 **Kieli** Englanti

Tiivistelmä

Työn tavoitteena oli selvittää käytännönläheisiä kuumalämpöpumppu (HTHP) tekniikoita ja niiden mahdollisia käyttökohteita lämpötiloissa ainakin 100 °C saakka lämmitystehoalueella 50…1000 kW sekä selvittää HTHP:jen kannattavuutta. Tekniikat ja käyttökohteet selvitettiin kirjallisuusselvityksellä. Selvitys keskittyi sekä tuoreimpaan tutkimukseen että käytäntöihin. Kannattavuutta selvitettiin Suomalaisen tapaustutkimuksen avulla. Selvityksen perusteella, kaupallisia HTHP malleja on ainakin 32 kappaletta, joista useat voivat tuottaa ainakin 100 °C lämpötilaa. Useissa teollisuuden sektoreissa on prosesseja, joiden lämmöntarvetta voitaisiin kattaa lämpötiloja 100 °C saakka tuottavilla HTHP:illa. Lisäksi, nykyiset kaukolämpöverkot ovat lupaava käyttökohde HTHP:ille, koska lämmöntarve on yleisesti alle 100 °C lämpötiloissa. Selvityksen perusteella, kylmäaine, kompressori ja kiertoprosessin kokoonpano ovat kriittisimmät huomioonotettavat seikat HTHP:illa. On sekä tieteellisiä ja käytännöllisiä merkkejä siitä, että HTHP-tekniikka on olemassa ainakin 100 °C lämpötiloille, jopa 165 °C lämpötiloihin saakka. Kylmäaine tulisi valita siten, että sen kriittinen lämpötila on vähintään 10…15 °C lauhtumislämpötilaa korkeampi ja höyrynpaineet pitäisivät olla sellaiset, että kompressori voi kestää ne, sekä imu- että painepuolella. Periaatteessa, kylmäainevalinta määrittelee korkeimman lämpötilan, jota voidaan tuottaa HTHP:lla, ainakin lämpötilaan 165 °C saakka. Korkeampien lämpötilojen suhteen, kompressorin suurin sallittu lämpötila (kuumakaasu), noin 180 °C, näyttäisi olevan rajoittava tekijä ollen kompressoritekninen haaste. Pääpiirteissään, lauhtumis- ja höyrystymislämpötilojen ero (lämpötilan nousu) määrittelee sen, että tulisiko käyttää yksi- vai kaksiasteista puristusta. Useimpien kylmäaineiden tapauksessa, sisäisen lämmönsiirtimen (IHX) käyttö on suositeltavaa minimitulistuksen tuottajaksi sekä, koska lämpökerroin ja lämmityksen tilavuustuotto voivat parantua ainakin 20 %. IHX parantaa saman verran HTHP:jen ympäristöystävällisyyttä ja energiatehokkuutta. Yleistäen, lämpötilan nousut 60 °C saakka voivat olla toteuttamiskelpoisia yksiasteisella puristuksella. Korkeammat lämpötilan nousut, jopa 120 °C saakka, pitäisi toteuttaa kaksiasteisella puristuksella välttääkseen suorituskyvyn heikkenemistä. Lisäksi, kaksiasteista puristusta voidaan käyttää alentamaan kuumakaasun lämpötilaa ollen erityisen tärkeää muutamien kylmäaineiden tapauksessa sekä käytettäessä IHX. Tapaustutkimuksen olosuhteiden mukaan, HTHP:t voivat olla tällä hetkellä keskimäärin 4,5 kertaa ympäristöystävällisempiä ja noin 1,5…5,5 kertaa energiatehokkaampia Suomalaiseen keskinkertaiseen kaukolämmitykseen verrattuna. Tapaustutkimuksen tapauksessa, HTHP:t ovat myös taloudellisesti kannattavia, koska investoinnin koroton takaisinmaksuaika olisi arviolta noin kaksi vuotta. Kuitenkin, kannattavuus on erittäin herkkä lähtötietojen suhteen.

Avainsanat kylmätekniikka, kuumalämpöpumppu, energiankierrätys, kylmäaine, COP

Preface

I made this thesis as the final part of my studies within the master's program in Advanced Energy Solutions in Aalto University by focusing to the major Sustainable Energy in Buildings and Built Environment. I made the research of this thesis during a period between December 2019 and March 2020. Both the client and the sponsor of this thesis was Chiller Oy. One day, I randomly checked advertised thesis topics via Aalto University sites where I noticed that this thesis subject was open. I decided to take this challenge because this subject seemed to be interesting and important due to an increasing demand as for the energy recycling to meet the set climate targets for the future from which perhaps the most binding is the Paris Agreement. All in all, I wish to thank both MSc Ari Aula and MSc Aleksi Aarnio who acted as the advisors from the Chiller Oy by sharing fruitful practical knowledge. Special thanks to professor Ville Vuorinen who advertised this topic and also acted as the supervisor.

Espoo 24.04.2020

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Contents

Nomenclature

Abbreviations

CAS	Cascade heat exchanger
CCHP	Combined Cooling Heating and Power
CHP	Combined Heating and Power
DC	Datacenter (including smaller-scale server rooms)
DH	District Heating
DHC	District Heating and Cooling
EEV	Electronic Expansion Valve
EU	European Union
GHG	Greenhouse Gas
GWP	Global Warming Potential
HP	Heat Pump
HTHP	High Temperature Heat Pump
HX	Heat Exchanger
IEA	International Energy Agency
IHX	Internal Heat Exchanger
IPCC	Intergovernmental Panel on Climate Change
ODP	Ozone Depleting Potential
ODS	Ozone Depleting Substance
TES	Thermal Energy Storage
TEV	Thermostatic Expansion Valve
VHTHP	Very High Temperature Heat Pump
VRE	Variable Renewable Electricity
VSD	Variable Speed Drive

Latin symbols

COP	Coefficient of Performance	-
COSP	Coefficient of System Performance	-
C	Cost of energy	€/kWh
h	Enthalpy	kJ/kg
I	Investment	€
i	Injected refrigerant mass flow rate	kg/s
$\dot{L}$	Leakage rate	kg/a
M	Molar mass	g/mol
m	Refrigerant charge	kg
$\dot{m}$	Refrigerant mass flow rate	kg/s
NBP	Normal Boiling Point	°C
n	System operational life	a
O	Annual operation time	h/a
p	Pressure	bar
$\dot{Q}$	Thermal power	kW
SC	Subcooling	°C
SH	Superheating	°C
SPP	Simple Payback Period	a
S	Annual saving	€/a

s	Entropy	kJ/kg°C
T	Temperature	°C
TEWI	Total Equivalent Warming Potential	kg-CO_2
$\dot{V}$	Volume flow rate	m^3/s
VHC	Volumetric Heating Capacity	kJ/m^3
$\dot{W}$	Electrical power	kW
W	Electrical energy consumption	kWh/a

Greek symbols

α	Recovery coefficient	0…1
β	CO_2-emission coefficient	kg-CO_2/kWh
Δ	Difference in subsequent parameter	-
ϵ	CO_2-emission of supplied heat	kg-CO_2/kWh
η	Efficiency	-
ρ	Density	kg/m^3

Subscripts and superscripts

alt	Alternative
boiler	Boiler
car	Carnot
comp	Compressor
cond	Condensation
cool	Cooling
crit	Critical
DH	District Heating
disp	Displacement
evap	Evaporation
el	Electrical
heat	Heating
high	Highest
in	Inlet
is	Isentropic
lift	Lift
lor	Lorenz
low	Lowest
max	Maximum
min	Minimum
out	Outlet
pinch	Pinch point
ratio	Ratio
ref	Reference
sat	Saturation
sink	Heat sink
source	Heat source
vol	Volumetric

Refrigerants (ASHRAE 34-2019 designation, R = Refrigerant)

CFC	Chlorofluorocarbon
R11	Trichlorofluoromethane
HCFC	Hydrochlorofluorocarbon
HFC	Hydrofluorocarbon
R134a	1,1,1,2-Tetrafluoroethane
R245fa	1,1,1,3,3-Pentafluoropropane
HCFO	Hydrochlorofluoroolefin (unsaturated hydrochlorofluorocarbon)
R1224yd(Z)	(Z)-1-chloro-2,3,3,3-Tetrafluoro-propene
R1233zd(E)	Trans-1-chloro-3,3,3-Trifluoro-1-propene
HFO	Hydrofluoroolefin (unsaturated hydrofluorocarbon)
R1234yf	2,3,3,3-Tetrafluoro-1-propene
R1234ze(E)	Trans-1,3,3,3-Tetrafluoro-1-propene
R1234ze(Z)	Cis-1,3,3,3-Tetrafluoro-1-propene
R1336mzz(E)	Trans-1,1,1,4,4,4-Hexafluoro-2-butene
R1336mzz(Z)	Cis-1,1,1,4,4,4-Hexafluoro-2-butene
HC	Hydrocarbon
R290	Propane
R600	Butane
R600a	Isobutane
R601	Pentane
R601a	Isopentane
R1270	Propene
Natural	
R717	Ammonia
R718	Water
R744	Carbon dioxide

1 Introduction

This thesis belongs to the refrigeration technology research field with a focus on heat pumps (HP) that can generate high temperatures. The purpose was to implement a literature survey on practical high temperature heat pump (HTHP) technologies and applications along with to investigate the profitability of HTHPs. This thesis was done for Finnish Chiller Oy (client) that is a subsidiary of Finnish Koja Group Oy. The client designs and manufactures solutions that meet cooling, heating and energy needs that are used in buildings and industry [1].

Immediate climate actions have been noticed to be essential because several warmest years have been recorded during the last two decades. Without accelerating international climate actions, the global average temperature increase is expected to reach 2 °C above the pre-industrial levels soon after 2060 and to continue increasing afterwards. [2] The Paris Agreement [3] aims to restrict the increase in the global average temperature to well below 2 °C above the pre-industrial levels and pursues efforts to limit the temperature increase to 1.5 °C. The European Union (EU) aims to be a climate-neutral by 2050 meaning an economy with net-zero greenhouse gas (GHG) emissions [2]. The Finnish government has announced to work such that Finland is a carbon neutral country by 2035 and a carbon negative soon after that. The Finnish target is announced to be implemented by accelerating emission reduction actions and strengthening carbon sinks. [4] Both the EU and the Finnish climate targets originate from the Paris Agreement [2,4]. As a one alternative to accelerate the emission reductions, HPs can be used to recycle energy and thus, reduce the use of the primary energy.

Figure 1.1 shows a Finnish energy consumption by sector (left) and in the industry (right) that were experienced few years ago. The energy use in the industry and space heating in buildings is together around three quarters from the total energy consumption. From the industrial energy use, fuels are mainly used to generate heat and electricity for a plant’s own use [5]. From the industrial energy use, around 40 % is covered with renewable wood fuels (biomass) and the remaining is covered in principle with fossil energy sources [6]. According to [7], biomass accounted for 33 and 75 % from the thermal energy consumption in district heating (DH) networks and in industry respectively in 2018 along with the remaining was mainly fossil fuels and peat. In the EU, around 50 % of total energy consumption was heating and cooling in 2015 [8]. The thermal energy potentials are such that HPs could generate depending on the demanded temperature levels.

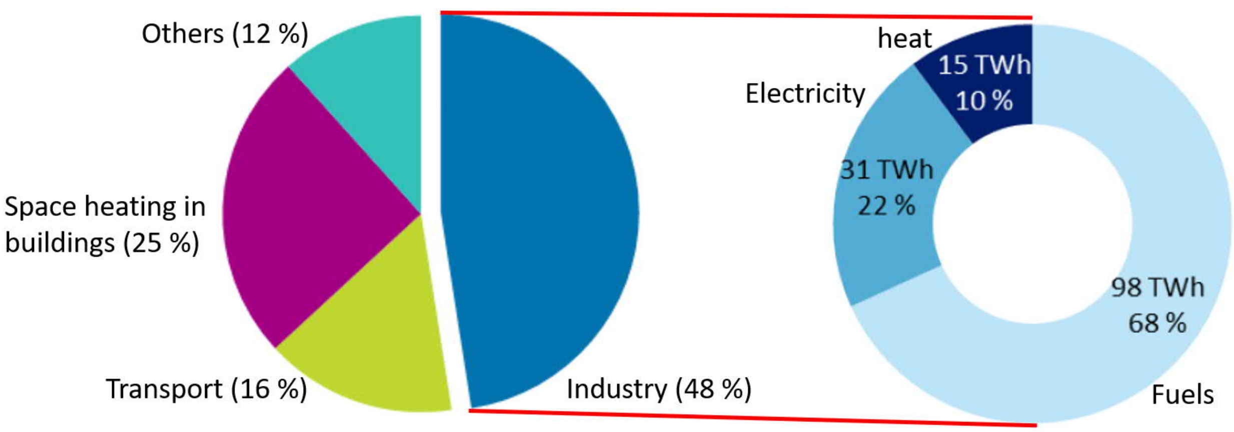

Figure 1.1. Energy consumption by sector in Finland (left) and energy consumption in the Finnish industry (right), combined from [6,9].

Thus, there are significant energy potentials that currently cause emissions and should be replaced with some emission-free solutions according to the mentioned climate targets. Currently, the biomass is considered both as a sustainable and as a renewable energy source at least in Finland and the EU [10]. As mentioned, the biomass is heavily used in the Finnish heat generation. However, increasing significantly biomass harvesting, e.g. forests, for energy use via combustion is widely criticized due to the regeneration of the biomass is slow and thus it can contribute the climate change [11].

To answer to the climate targets, recycling heat with electricity-driven HPs can be considered as a one solution for the carbon neutral heat supply if they are operated by using electricity from renewable energy sources [12]. It is expected that electricity prices can, at least intermittently, decrease in the future due to an increase in a variable renewable electricity (VRE, e.g. wind) generation [12-14]. The increase in the VRE can be necessary due to the climate targets. This means that HPs are expected to become environmentally more friendly, more energy efficient and economically more profitable along with e.g. combined heating and power (CHP) plants can become less profitable [12].

In Finland, HPs are well established technology, especially in the residential sector as shown in Figure 1.2 where cumulative small-scale residential HP sales indicate that around one million residential HPs were installed in Finland by 2020. Despite of this, it was noticed during the literature survey that HPs are not commonly used either in larger scale installations or in the industry.

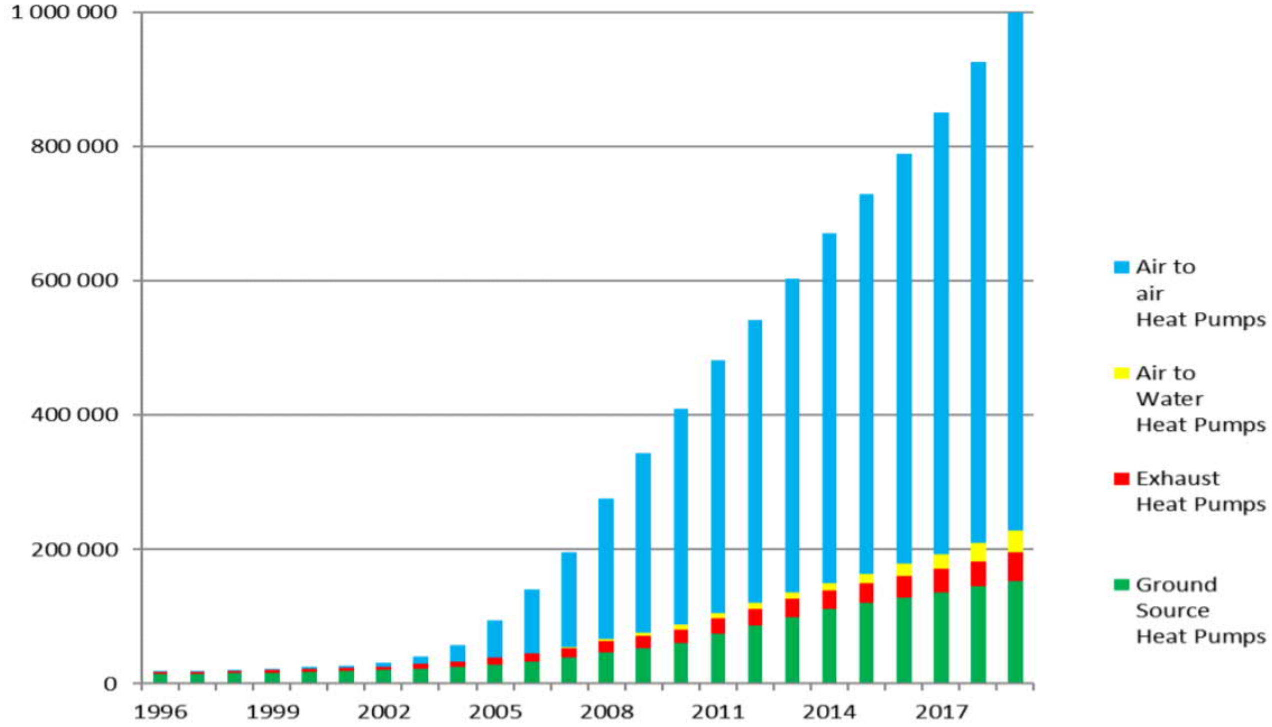

Figure 1.2. Cumulative small-scale residential heat pump sales in Finland by 2020 [15].

HPs can be used to recycle heat by recovering a low temperature waste heat and upgrading it to a higher temperature which is required in a process. Figure 1.3 shows the energy flows and the basic cycle configuration of the electricity-driven closed-cycle vapor compression HP (later in this thesis: ‘’basic cycle’’). As the refrigerant (working fluid) is circulated with a compressor, it recovers heat from the heat source before it is compressed to a higher temperature and pressure at which it supplies the heat to the heat sink and further, it is being throttled back to the lower temperature and pressure. This cycle process is being repeated with HPs. As for vapor compression HPs, the refrigerant is evaporated and possibly superheated when heat is recovered from the heat source along with desuperheated, condensed and subcooled when supplying heat to the heat sink. So, HP upgrades a certain amount of heat from the heat source $\dot{Q}_{source}$ at a lower temperature T_{source} to a higher temperature T_{sink}. For bringing the heat to the higher temperature, a certain proportion of electricity $\dot{W}$ is needed to operate the compressor. If no heat losses occur to the environment, the amount of heat supplied to the heat sink $\dot{Q}_{sink}$ is the sum of the heat from the heat source and the electricity used to operate the compressor. HPs enable waste heat recycling and are often used to replace combustion based heat supply, thus contributing to an improved energy efficiency. [16]

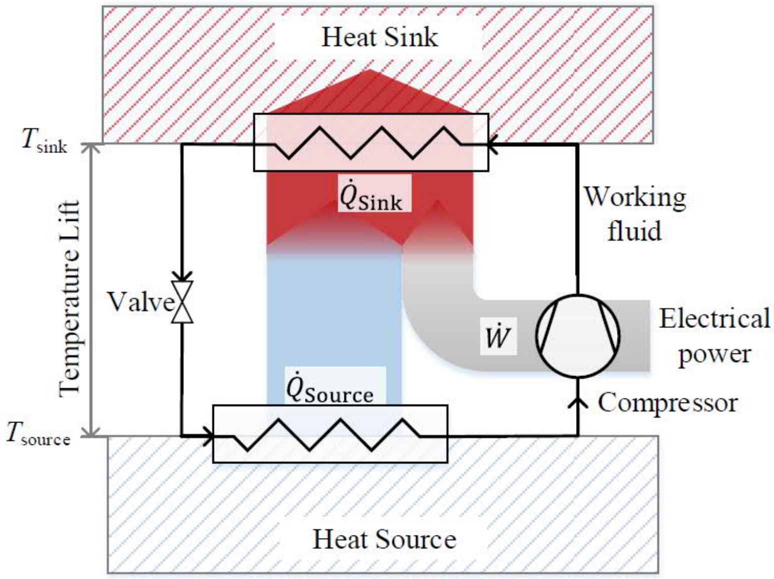

Figure 1.3. Basic vapor compression heat pump cycle and its energy flows [16].

The term ''waste heat'' is an unused part from an energy input to a process or processes. The waste heat is often at a low temperature and is discharged to the environment because it cannot be used in processes or in alternative processes. [5] In this thesis, the heat sources are limited to the waste heat from the industry and the built environment.

The heat sources are limited to waste heat sources since the purpose is to survey such HTHP technologies and applications where waste heat can be recycled and thus, used again in a useful temperature level. Let us consider a process where both the waste heat source and the heat sink are linked to a same process that is running all the time. In this process, a limitless recycling process can be considered since the waste heat from the process could be steadily upgraded to the higher level and the useful heat could be used in the process and so on. In practice, a certain proportion of electricity is needed to bring the waste heat from the lower to the higher temperature level and the amount of temperature rise from the lower to the higher temperature level is limited. HTHPs operated with VRE could be promising since the whole described process could be in principle a carbon neutral. However, e.g. a current Finnish average electricity mix in the grid is not a carbon neutral, but a current target is to decrease the emissions towards zero, at least very close to the zero [10]. Currently, the carbon emissions from fossil fuels are around two times higher when compared to the electricity in Finnish grid [10]. Even currently, the carbon emissions can be significantly reduced when comparing to the fossil fuels by using HTHPs because only a certain proportion of the heat supplied to a heat sink is electricity.

Figure 1.4 classifies HPs by temperature levels. According to [17], HPs for up to 80 °C heat sinks are conventional HPs that are established in buildings and in the industry, HTHPs are currently commercially available up to around 100 °C heat sinks along with very high temperature heat pumps (VHTHP) are currently in a development phase, especially for over 140 °C heat sinks, that are mainly laboratory scale research systems. The term HTHP is often used in connection with industrial heat pumps that are mainly used in the industry to recycle heat [17]. The industrial heat pumps use the waste heat as the heat source and deliver heat at a higher temperature for use in industrial processes and space heating along with can provide industrial cooling [18]. However, the term HTHP is not consistent in the literature since the lower temperature limit for the heat sink varies between 80 °C and 100 °C while the industrial heat pumps have been defined for heat sinks up to 150 °C [17,19].

Further in this thesis, only HTHPs that can provide heat sinks up to between 80…100 °C are considered unless else mentioned. Moreover, this thesis does not distinguish between HTHP and VHTHP if higher temperatures than 100 °C are discussed. And, the term HTHP is defined such that it can deliver at least 80 °C heat sinks along with the applications for the industrial heat pumps are considered by expanding them with the DH networks as the potential application.

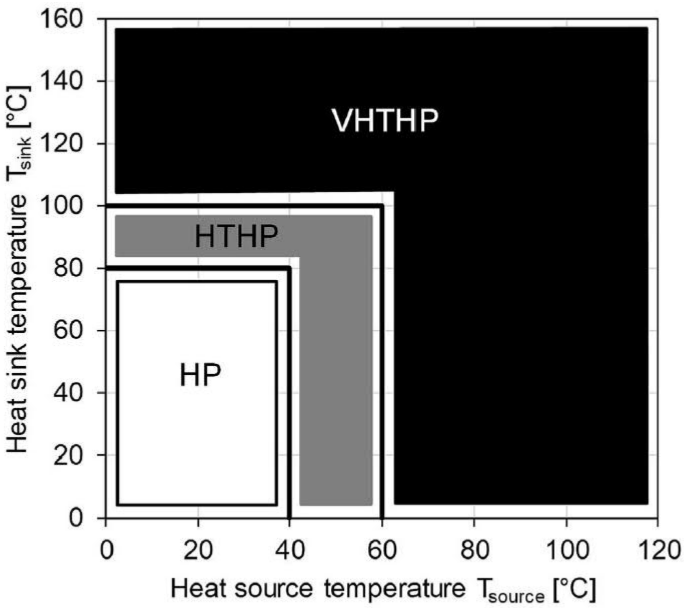

Figure 1.4. Classification of heat pumps by temperature levels [17].

The rest of this thesis is organized as follows: section 1.1 determines the objectives and the limitations of this thesis and section 2 discusses methodologies of this survey. Section 3 goes through the relevant HP theory. Furthermore, current commercialized HTHPs are summarized in section 4 and section 5 discusses potential HTHP applications followed by section 6 that discusses technical special solutions that are required for higher temperatures and section 7 that presents practical state-of-the-art HTHP technologies. In section 8, a Finnish case study is implemented to estimate the profitability of HTHPs. Finally, section 9 concludes.

1.1 Objectives and limitations

In this section, research objectives and limitations of this thesis are determined. Both the objectives and limitations were gathered from the client.

The objectives were to find out:

- Commercialized HTHPs in the world currently (section 4);
- Potential HTHP applications and their needs (section 5);
- What special technical solutions are needed for the higher temperatures (section 6);
- Practical HTHP technologies (section 7);
- Profitability of HTHPs via a case study (section 8).

Limitations for the applications, i.e. where HTHPs could be applied, were determined as below.

- Application needs consists of waste heat and demanded heat sink temperature levels along with energy quantities related to these;
- Application energy quantities are surveyed exemplary only from Finland;
- Heat sources are limited to the waste heat, thus e.g. geothermal heat and ambient air are not considered;
- Only the industry and large buildings are considered.

Limitations for the practical HTHP technologies were determined as below.

- Only closed-cycle vapor compression technologies are considered;
- Only electricity-driven compressors are considered;
- Heat sinks up to between 80…100 °C must be at least achieved;
- Heating capacity from 50 to 1000 kW must be achieved with one HTHP;
- Only pure one-component refrigerants are considered;
- Only subcritical cycles are considered;
- Refrigerant and main components of HTHP are considered and from the main components, practical technologies for the compressor are only surveyed.

According to the client, there was no need to find out e.g. practical heat exchangers that are used e.g. as condensers and evaporators along with expansion valves since basic components that are suitable for the conventional HPs could be in outline suitable for HTHPs as well. On the other hand, especially the refrigerant and the compressor were considered to be such components that require special technical solutions to expand the temperature range from the HP to the HTHP level.

2 Methodology and the most significant references

For the most part, the implementation of this thesis was based on a literature survey that consisted of finding out practical HTHP technologies and potential applications for HTHPs. In addition to the survey, the relevant theory and other matters were found out from the current literature and handbooks. As for the survey, several sources were utilized to find out the most promising references. The most significant databases were as follows: research papers from ScienceDirect via the Aalto University, International Energy Agency (IEA) conference papers from the home pages of the IEA Heat Pumping Technologies, official research database of the Technical University of Denmark (DTU Orbit), conference papers from ResearchGate and as for a general search, Google Scholar. In addition, available data from manufacturers' home pages were surveyed to find out the current commercialized HTHPs, state-of the-art synthetic HTHP refrigerants and potential compressors.

The reliability of the databases was evaluated to be sufficient. According to ScienceDirect home pages, they host over 16 million research papers along with they serve academic institutions, governments, companies across all industries and millions of researchers. According to the home pages of the IEA Heat Pumping Technologies, they have been active around 40 years, they participate with the government of countries and have 17 member countries along with organizes a conference every third year (the IEA Heat Pump Conference) whose purpose is to bring together the world's leading industrial, scientific and political actors to discuss the latest advancements in the field of the heat pumping technologies and their applications. As for the DTU Orbit, it was considered to be a reliable source since authors that were present in papers that were found from the DTU Orbit, were also present in papers that were hosted by ScienceDirect and IEA conferences. According to the home pages of ResearchGate, it has over 16 million users along with this database was used to assist in finding suitable papers resulting that several conference papers were found that were mainly from the same authors that were present in the previous databases. In addition, Google Scholar was mainly used to gather the total number of citations for the found papers along with it was noticed that it in outline found the same papers that were present in ScienceDirect and IEA conferences.

The quantity of the available research papers seemed to be sufficient for the implementation of the survey. Even, the quantity of the papers seemed to be so large that all of those could not be surveyed in the light of the man-hours reserved for this thesis. Thus, there was a need to select the most promising papers that were utilized in this thesis. The quality of the papers was monitored like discussed in the next paragraph. There was two major ways to monitor the paper quality from which the other was based on the publication year and the number of citations per a year along with the other was based on that how often the authors of a certain paper are present in all of the found papers. The monitoring caused that all of the potentially suitable papers were not utilized in this thesis.

The papers that were finally utilized to find out the practical HTHP technologies and potential applications are evaluated in Figure 2.1, where only research papers hosted by ScienceDirect along with conference papers are listed. The citations were gathered from Google Scholar 9th March 2020. In Figure 2.1, the papers are evaluated according to the paper publication year along with the number of citations per a year for a certain paper. As shown, 66 scientific research and conference papers were utilized for the survey from which the majority are published in 2014…2019 indicating that the survey was based on the current research and practices. The number of citations per a year shows that there is a significant deviation. On average, the annual citation number is 6 pieces. But, some papers have even around 15…20 citations per a year and the reference from the year 2013 has even 70 citations per a year (561 pieces in total). However, there are several papers that have a relatively low number of citations. However, it was considered that those can be utilized in conjunction with other papers to gain more evidence on the findings. In addition, the lack of the citations was not considered a major issue in several cases since the authors of a certain low-cited paper were noticed to be the authors in other papers that were cited more. In this way, the quality of the selected papers was monitored, too. Along the survey, significant similarities along with cross-citing were noticed across the utilized papers that gave even more evidence on both the reliability of the papers and their findings. Furthermore, popular citations occurring in the papers were utilized within the survey to find more promising papers.

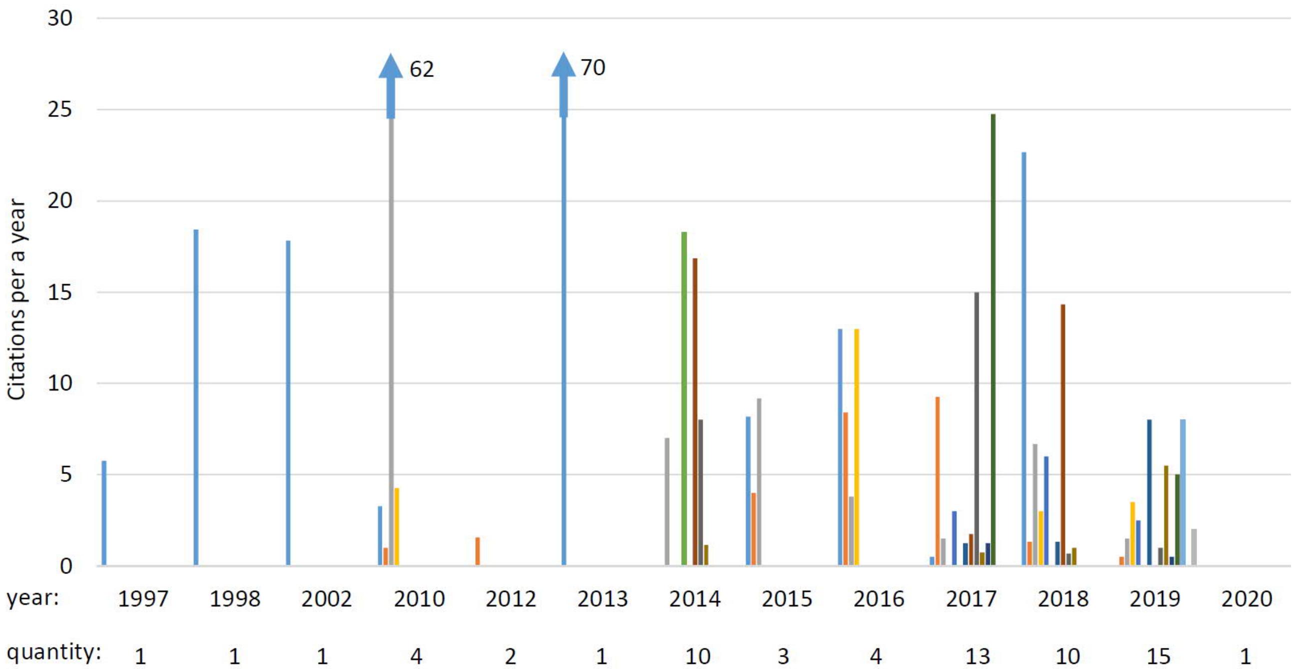

Figure 2.1. Utilized scientific research and conference papers for the survey on the practical HTHP technologies and potential applications.

It was considered that the utilized papers should both be significant for this thesis and have an enough high citation intensity. As shown in Figure 2.1, there were even 66 papers solely from scientific databases that were significant for this thesis. In addition to those, numerous other sources were used, such as manufacturers' home pages along with reports from some known actors. As for an informative summary for the most significant papers that were utilized for the survey on the practical HTHP technologies and potential applications, see Appendix 1.

The methodology to find out the profitability deviated from the rest of this thesis because the profitability of HTHPs was found out via a case study. The profitability was estimated with respect to three different aspects that were the economics, the friendliness to the environment and the energy efficiency. Moreover, the sensitivity of the profitability was estimated to see the potential boundaries of the profitability when considering an uncertainty in the source information.

3 Heat pump

The main purpose of this section 3 is to go through the relevant theory concerning this thesis along with to explain in detail what a HP is and how it works. Moreover, the main components of a HP will be discussed to explain their purpose and how those can influence the performance. In addition, the limitations that are not handled further in this thesis are discussed in suitable sections. The term ''HP'' is used for the generalization although e.g. a refrigeration unit works with the same principle. Through this thesis, the refrigerants are considered to be pure one-component substances along with the cycles are considered to be subcritical unless else mentioned. In addition, the expansion devices are considered to be expansion valves in this thesis.

HPs can be classified according to that is the system closed or open [17]. In the closed systems, the working fluid (refrigerant) is circulated to transfer heat such that the same fluid stays inside the system. These systems consist of vapor compression and absorption systems. On the other hand, the open systems are such where the working fluid is continuously replaced when the process is running. These systems consist of mechanical vapor recompression and thermal vapor recompression systems. The most common working fluid in these open systems is water. [18,20] The closed-cycle vapor compression is the most common HP technology [17,21] and is in the focus within this thesis.

3.1 Basics

In this section, the basic cycle illustrated earlier in Figure 1.3 is explained in detail. This cycle is sometimes referred as a reversed Carnot cycle because the original concept was a heat engine generating work in a power generation cycle operating clockwise [22]. Secondly, relevant equations to assess HP performance will be presented. Thirdly, the cycle process is classified according to the refrigerant working domain and the critical point. Fourthly, multistage cycles are discussed.

3.1.1 Cycle process

In theory, the basic cycle is illustrated in Figure 3.1 with the cycle configuration having the main components and log(p)-h-diagram where the refrigerant state points are numbered from 1 to 4. On the right-hand side in Figure 3.1, the refrigerant dependent log(p)-h-diagram is an essential in the refrigeration technology to understand the cycle process. The diagram pressure is absolute. The refrigerant is subcooled liquid on the left-hand side of the saturated liquid curve, whereas it is superheated vapor on the right-hand side of the saturated vapor curve. Over the critical point, the vapor cannot be liquefied. At the critical point, the critical pressure and the critical temperature are located. Isotherms are lines where the refrigerant temperature is a constant. The isotherms are vertical lines at the liquid phase, horizontal lines between the saturation curves (excluding zeotropic mixtures) and curve to down at the vapor phase. Isentrops are lines where the entropy is a constant. Isochors are not shown in Figure 3.1, but those are a constant density curves that are rising from the liquid to the vapor phase. [23] The refrigerant vapor pressure is defined at the intersection of the saturated vapor curve [24].

As for the subcritical closed-cycle vapor compression HPs, the process is based on a cycle where the circulating refrigerant evaporates and condenses. In this cycle, HP main components are the evaporator, the compressor, the condenser and the expansion valve. In the following, changes from a state point to a next state point shown in Figure 3.1 are explained according to [22,23].

Evaporation (4→1)

In the evaporator, the refrigerant absorbs heat from the heat source by evaporating in a temperature that is lower than that of the heat source. The refrigerant enthalpy increases. The evaporation occurs in a constant pressure. The refrigerant is at least slightly superheated because the liquid phase present in the refrigerant during the compression can damage the compressor.

Compression (1→2)

The compressor sucks a low pressure superheated vapor refrigerant and compresses it to a higher pressure and at the same time, the vapor temperature rises. The temperature at the compressor suction is often called as a compressor suction temperature and the temperature after the compression as the compressor discharge temperature. The compression shown is not isentropic because a certain entropy generation occurs. The theoretical isentropic compression could have a constant entropy during the compression.

Condensation (2→3)

In the condenser, the refrigerant supplies heat to the heat sink by condensing in a temperature that is higher than that of the heat sink. The refrigerant enthalpy decreases. In the condenser, the refrigerant gets first desuperheated before the saturated vapor curve, secondly gets condensed between the saturation curves and thirdly gets subcooled after the saturated liquid curve. Subcooling ensures that refrigerant is 100 % liquid before the expansion valve. If some vapor is present before the expansion valve, it can cause an excessive pressure loss and thus reduction in HP performance.

Expansion (3→4)

In the expansion valve, the pressure of the liquid refrigerant reduces and thus, the liquid changes to a two-phase liquid-vapor and at the same time, its temperature reduces to the evaporation temperature.

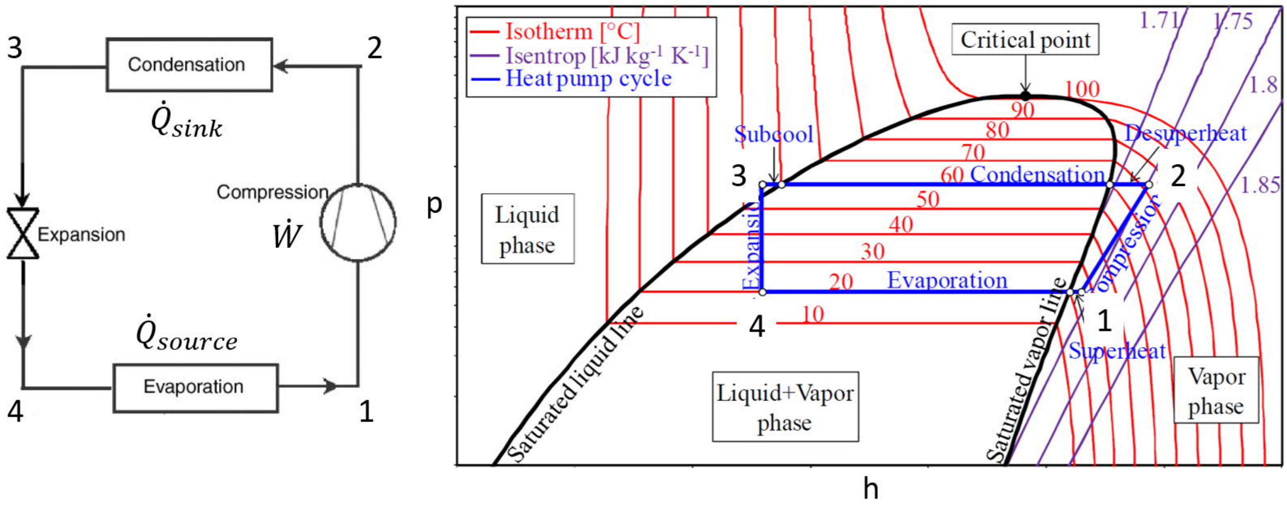

Figure 3.1. Cycle configuration (left) and log(p)-h-diagram (right) of the basic cycle (modified from [22,25]).

In practice, there occurs following losses that were not either discussed or shown in Figure 3.1. Pressure losses occur in the evaporator, in the condenser and in the piping. Also, heat losses occur. These causes that the pressure is decreasing from 4 to 1 along with from 2 to 3. Moreover, the pressure losses occur in the compressor. Heat losses occur during the compression which leads to that the compression is not isentropic. Note that the non-isentropic compression leads to that the line from 1 to 2 curves more to the right meaning larger compressor electricity consumption and higher compressor discharge temperature when compared to the theoretical isentropic compression. [23]

3.1.2 Performance indicators

In the following, relevant equations are presented. These are commonly used to determine the cycle performance of HPs. A one common indicator is shown in Equation 1 that is the coefficient of performance (COP), which relates the supplied heat $\dot{Q}_{sink}$ to the compressor electricity consumption $\dot{W}$ [16,26]. The numbering is according to Figure 3.1.

$$COP = \frac{\dot{Q}_{sink}}{\dot{W}} = \frac{\dot{m} \cdot (h_2 - h_3)}{\dot{W}} = \frac{\dot{V}_1 \cdot \rho_1 \cdot (h_2 - h_3)}{\dot{W}} \tag{1}$$

The COP shown in Equation 1 is the COP for heating. The heating COP is solely used in this thesis unless else mentioned. If a HP is simultaneously utilized to supply a cooling demand by cooling down a heat source and to heat up a heat sink, COP can be calculated as shown in Equation 2. [27,28]

$$COP_{cool+heat} = \frac{\dot{Q}_{source} + \dot{Q}_{sink}}{\dot{W}} \tag{2}$$

The maximum achievable COP of a HP is determined by the temperatures of the heat source and sink between which a HP is working. The COP is only suitable for comparing HPs when evaluated for the same heat source and sink temperatures. There are basically two ways to determine the maximum achievable COP. To determine the COPs in Equations 3 & 4 below, the temperatures are in kelvins. [16]

For the conventional Carnot cycle, a HP operates between a heat sink and a heat source that are at constant temperatures T_{high} and T_{low} as shown in Equation 3. The HP absorbs and supplies heat at constant temperatures along with an infinite heat source and sink are assumed. In addition, the Carnot cycle assumes that there is no temperature difference between the refrigerant and the heat source along with between the refrigerant and the heat sink, hence infinite heat exchanger areas for the evaporator and the condenser are assumed. Thus, the HP has to operate between the lowest and highest temperatures $T_{source,out}$ and $T_{sink,out}$. [16]

$$COP_{car} = \frac{T_{high}}{T_{high} - T_{low}} = \frac{T_{sink,out}}{T_{sink,out} - T_{source,out}} \tag{3}$$

If the heat source and the heat sink experience a temperature difference during the heat transfer (temperature glide) with the refrigerant, a certain inefficiency occurs in transferring heat over a temperature difference. The Carnot cycle is not optimal within these cases. The Lorenz cycle operates similarly between the heat source and the heat sink. However, the heat transfer of the Lorenz cycle occurs at the average temperatures of the heat source $\bar{T}_{source}$ and the sink $\bar{T}_{sink}$ as shown in Equation

4. The average temperatures can be approximated by using the logarithmic mean temperatures. The Lorenz cycle is optimal for heat sources and sinks that experience a temperature glide. [16]

$$COP_{lor} = \frac{T_{high}}{T_{high} - T_{low}} = \frac{\bar{T}_{sink}}{\bar{T}_{sink} - \bar{T}_{source}} = \frac{\dfrac{T_{sink,out} - T_{sink,in}}{\ln\left(\dfrac{T_{sink,out}}{T_{sink,in}}\right)}}{\dfrac{T_{sink,out} - T_{sink,in}}{\ln\left(\dfrac{T_{sink,out}}{T_{sink,in}}\right)} - \dfrac{T_{source,in} - T_{source,out}}{\ln\left(\dfrac{T_{source,in}}{T_{source,out}}\right)}} \tag{4}$$

The Carnot efficiency relates COP to COP_{car} as shown in Equation 5. The Lorenz efficiency relates COP to COP_{lor} analogously to the Carnot efficiency. These efficiencies give a feeling about how high the COP of the actual HP is when compared to the maximum achievable COP. [16]

$$\eta_{car} = \frac{COP}{COP_{car}} \tag{5}$$

Additional case-specific indicators can be derived by using parameters from the cycle process. The volumetric heating capacity (VHC) is defined by the ratio of heat supplied to the heat sink per compressor suction refrigerant volume flow rate as shown in Equation 6. The higher VHC, the better since a smaller compressor could be needed for a given heating capacity. The VHC is a commonly used performance indicator that gives a feeling from a needed compressor size and thus the size of a HP. Larger compressors needs more electricity and have higher heat losses with a given heating capacity. The VHC influences the HP investment since the compressor is a major part of the investment. [16,17,25,29] Numbers in Equation 6 refer to Figure 3.1.

$$VHC = (h_2 - h_3) \cdot \rho_1 = \frac{\dot{Q}_{sink}}{\dot{V}_1} \tag{6}$$

The temperature lift T_{lift} (Equation 7) is a commonly used and an important indicator due to it has a major impact on the maximum achievable COP. The higher T_{lift}, the lower is the maximum achievable COP (see Equations 3 & 4). [16,17] Due to the area of heat exchangers is limited, the condensation temperature T_{cond} must be higher than $T_{sink,out}$ if neglecting potential heating impact of desuperheating along with the evaporation temperature T_{evap} must be lower than $T_{source,out}$ [16]. In the light of the practical considerations, T_{lift} was determined to be the temperature difference between the condensation and the evaporation temperatures. As the pure refrigerants are only considered in this thesis, there is no need to consider the average temperatures in the definition of T_{lift}. In the literature, T_{lift} is determined both in this way and in a way that it is $T_{sink,out} - T_{source,out}$, i.e. according to the theoretical maximum COP [17,25].

$$T_{lift} = T_{cond} - T_{evap} \tag{7}$$

3.1.3 Cycle types

The cycles can be classified to three different types according to how the critical point of the refrigerant is located with respect to the cycle working domain (blue tetragon) as shown in Figure 3.2. The cycle is subcritical if both the heat collection and the supply occurs below the critical point. The cycle is transcritical if the heat collection occurs below the critical point while the heat supply occurs over the critical point. The cycle is supercritical if both the heat collection and the supply occurs over the critical point. The implication for the heat supply in the transcritical cycles is that the refrigerant experiences a temperature glide, whereas in the supercritical cycles, the refrigerant experiences a temperature glide during both the heat collection and the heat supply (see the isotherms). [25]

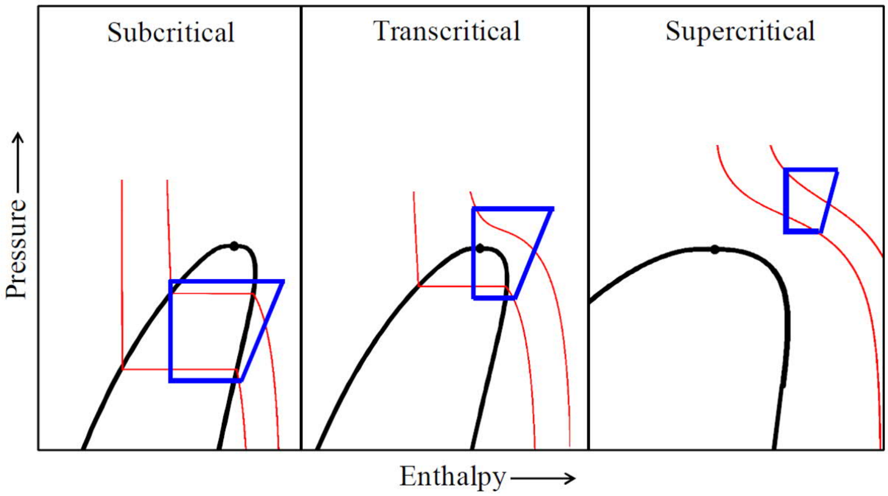

Figure 3.2. Subcritical, transcritical and supercritical cycles [25].

3.1.4 Multistage cycles

The multistage cycles are such cycles where the compression is implemented in two or more stages. If the ratio of compressor discharge to suction pressure (pressure ratio), practically T_{lift}, is too high for a 1-stage compression to cause a too high decrease in the compressor volumetric and isentropic efficiencies or an unacceptable high compressor discharge temperature, the compression can be implemented by using a multistage cycle, typically a 2-stage cycle that utilizes the same refrigerant through the whole cycle. The 2-stage cycles can apply either 2 separate compressors or a single compound compressor where the compression occurs in 2 stages. As T_{lift} is high enough and keeps increasing, the benefit from a multistage cycle increases in comparison to a 1-stage cycle. [22,23,30,31] The main principle in the multistage cycles is that the refrigerant from the previous compression discharge is intercooled before being sucked to the next compression. This intercooling can be implemented in several ways from which a one energy efficient option is presented next. [22]

Figure 3.3 shows a 2-stage cycle where the intercooling is implemented with an economizer circuit using an additional subcooler. A proportion of the condensed liquid mass flow i (blue circuit: a boundary 3,4,5,8) is expanded (pressure and temperature decreases) through an expansion valve (5→8) and flows through the subcooler (8→9) where it gets superheated by absorbing heat from the main circuit mass flow $\dot{m}$ (black circuit: a boundary 1,4,6,7) that gets more subcooled (5→6). The superheated vapor is then injected at the intermediate pressure to cool the main circuit flow between the compression stages (2→3). The compressor discharge temperature could be higher with one compression since the whole compression line (1→4) could curve more to the right. The additional mass flow i in the condenser along with the increased enthalpy difference for $\dot{m}$ in the evaporator (5-7) implicates that both COP and VHC increases when compared to the basic cycle. [22,32]

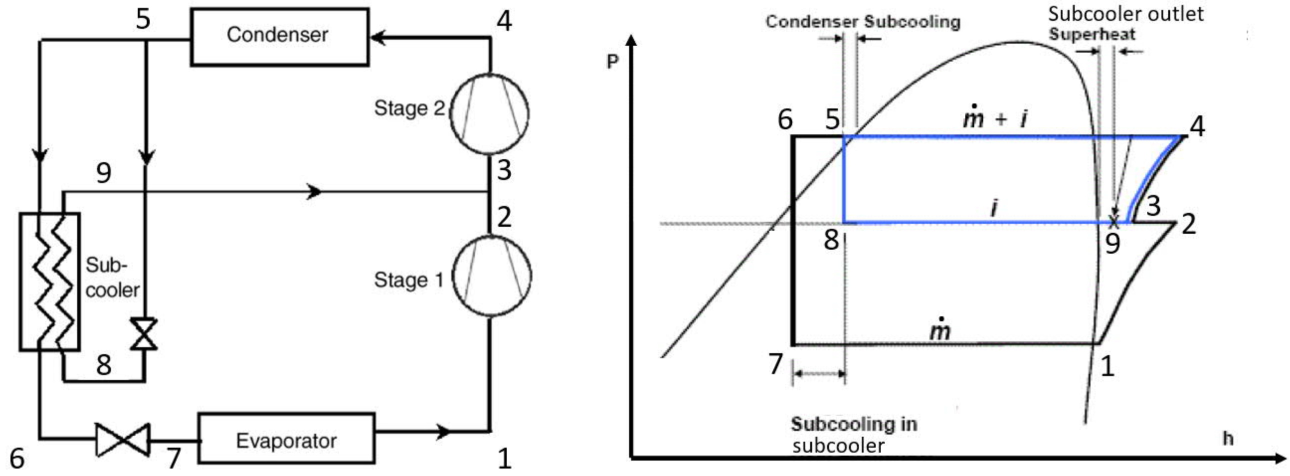

Figure 3.3. Cycle configuration and log(p)-h-diagram of 2-stage economizer cycle [22,32].

3.2 Refrigerants

This section states properties of an ideal refrigerant along with shortly describes refrigerant mixtures. Further, potential environmental impacts from refrigerant leakage and operation of HP are discussed. In the literature, the refrigerant is often called as ‘’working fluid’’. However, the term refrigerant is used in this thesis. On the other hand, use of the term ‘’working fluid’’ could be justified as well due to it describes the crucial technical meaning well like discussed in touch of the section 3.1.

3.2.1 Ideal properties

There are several properties that make a distinction between different refrigerants. Ideally, a refrigerant should have properties that are listed below. [17,22]

- Critical temperature well higher than condensation temperature to allow subcritical cycles;
- Critical pressure that is not excessive;
- Pressure at system standstill over 1 atm;
- High COP and VHC;
- Non-ozone depleting, i.e. ozone depleting potential (ODP) practically zero;
- As low global warming potential (GWP) as possible, preferably less than 10;
- Non-corrosive;
- Non-toxic;
- Non-flammable;
- Chemically stable;
- Compatibility with materials such as aluminum, steel and copper;
- Miscible with lubricants;
- Available on the market and with a low price.

However, there are not such refrigerants that satisfies all those listed properties. Thus, the refrigerant choice for any particular application will be always a compromise. [22]

3.2.2 Mixtures

Many of e.g. hydrofluorocarbons (HFC) are mixtures of two or more individual refrigerants. Mixtures are typically developed to achieve the best compromise in performance, GWP and flammability. [22] According to [16,23], there are two types of mixtures by nature like listed below.

- An azeotropic mixture behaves like a pure refrigerant, thus does not experiences a temperature glide during a phase change. These could be used as drop-in and replacement refrigerants.
- A Zeotropic mixture experiences a temperature glide during a phase change. If these are used, a new system design should be implemented to exploit the whole efficiency improvement potential when compared to the pure refrigerants and the azeotropic mixtures.

There are also some so called ‘’near azeotropic’’ mixtures that experience a small glide of less than 2 °C [22]. The Zeotropic mixtures can increase the cycle performance because the inefficiencies due to the heat transfer can be reduced when matching the heat source and the heat sink temperature glides to the refrigerant evaporation and condensation temperature glides [16]. The potential performance improvement from the use of the zeotropic mixtures is illustrated in Figure 3.4 showing that an effective T_{lift} is lower when using the zeotropic mixture due to the average temperatures. Thus, the maximum achievable COP is higher with the mixture that behaves according to Equation 4 when compared to a pure refrigerant whose maximum achievable COP behaves according to Equation 3.

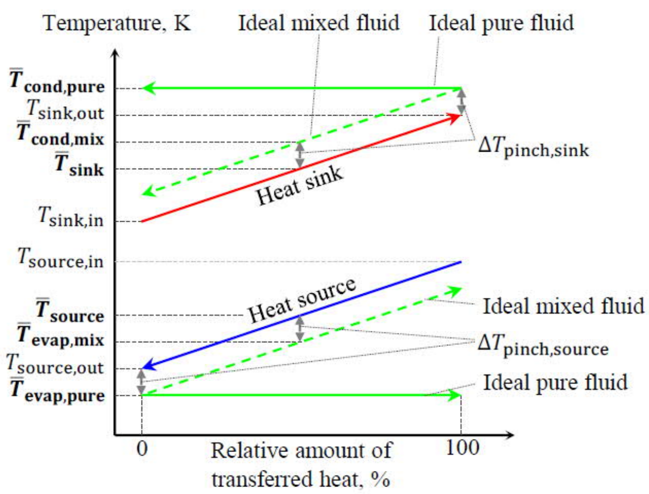

Figure 3.4. Temperature-heat-diagram of the basic cycle operating either with an ideal pure refrigerant or with an ideal zeotropic mixture refrigerant [16].

3.2.3 Environmental impact and legislation

To assess potential environmental impacts from the refrigerant emissions and operation of HPs, following three important indicators have been established.

ODP (Ozone Depleting Potential)

The ODP refers to the amount of ozone depletion caused by a refrigerant emission expressed as the ratio of a global ozone loss due to a given refrigerant and the global ozone loss due to R11. The ODP of R11 is a unity along with the ODP values of different refrigerants range from 0 to 10. The higher ODP, the more the refrigerant depletes the ozone layer. [33]

GWP (Global Warming Potential)

The GWP compares the global warming impact from a direct emission of a GHG relative to emitting the same amount CO_2 (R744) to the atmosphere, integrated over a fixed time horizon [34,35]. In this thesis, the GWP values integrated for a 100-year period are used like determined in [34].

TEWI (Total Equivalent Warming Potential)

The TEWI is a way of assessing the overall global warming impact by combining the direct emissions from the refrigerant leakage to the atmosphere with the indirect emissions from the electricity consumption to operate a HP system over its operational life. The TEWI can be calculated according to Equation 8 as described in [36], where the first term on the left is the impact of the refrigerant leakage, the second is the impact from losses in the refrigerant recovery and the third is the impact from the electricity consumption to operate the HP. [36] Currently, the major part from the global warming impact is typically due to emissions from the electricity to operate the HP. A European average GHG emission from the electricity generation is currently relatively high and when combined with the HP electricity consumption, the direct emissions from the refrigerant leaks are typically clearly smaller, even if the refrigerant GWP is high being e.g. 1000. [22,37]

$$TEWI = GWP \cdot \dot{L} \cdot n + [GWP \cdot m \cdot (1 - \alpha)] + n \cdot W \cdot \beta \tag{8}$$

Depending on the refrigerant, it has or does not have ODP and is more or less effective GHG if it is emitted to the environment [35]. Ozone depleting substances (ODS) cause stratospheric ozone layer depletion that is harmful because there, the ozone layer protects living beings in the world from the harmful ultraviolet radiation from the sun and further, an increased ultraviolet radiation has adverse effects on the ecosystem and on the human health by e.g. increasing the incidence of skin cancers [33]. On the other hand, the GHG emissions cause global warming [2]. Several refrigerants are even thousands of times more effective GHGs than the CO_2 [35].

As long as the refrigerant is inside the system (e.g. HP) and is salvaged in touch with breaking up the system along with is delivered for an appropriate disposal, the refrigerant does not harm the environment [23]. However, as for stationary systems, the refrigerant has been experienced to leak into the environment from 3 to 22 % from the system refrigerant charge annually. With appropriate actions, an annual leakage rate of less than 5 % is achievable. The combined direct CO_2-equivalent emissions from the leakage of chlorofluorocarbons (CFC), hydrochlorofluorocarbons (HCFC) and HFCs derived from an atmospheric observations decreased from around 7.5 $GtCO_2$-equivalent per year in 1990 to around 2.5 $GtCO_2$-equivalent per year in 2000 that was around 33 and 10 % respectively when compared to the annual CO_2-equivalent emissions due to global fossil fuel combustion. Reductions in the direct GHG emissions from the refrigerants can be achieved by improving the containment, reducing the charge, the end-of-life recovery and by recycling or by improving the disposal along with by using alternative refrigerants with a reduced or a negligible GWP. [35]

There have been several changes in the selection and the use of the refrigerants in time. Figure 3.5 shows the refrigerant development from the early times of the refrigeration technology to the current state. The first generation of refrigerants included whatever worked and was available. Accidents were common. Nearly all of these were flammable, toxic or both along with some were also highly reactive. The second generation of refrigerants was distinguished by a shift to fluorochemicals for safety and durability. Dominating refrigerants were CFCs whose commercial production began in the 1930s followed by HCFCs in the 1950s. R717 (ammonia) continued and remains currently, the most commonly used refrigerant in large industrial systems, especially in the food and beverage industry. The third generation of refrigerants was focusing to substitute CFCs and HCFCs to protect the stratospheric ozone layer due to those were noticed to be ODSs. Both CFCs and HCFCs were agreed to be phased out from which CFCs faster. HCFCs were used as transitional options while HFCs as longer alternatives. The trend sparked a renewed interest in natural refrigerants, especially hydrocarbons (HC), R717, R718 (water) and R744. Currently, the fourth generation of refrigerants focuses on mitigating the global warming. The intergovernmental panel on climate change (IPCC) has a scientific consensus on that the global warming is very likely due to the observed increase in the GHG concentrations. The issues on the global warming have led e.g. to that, GWPs smaller than 150 are only allowed in new automobile air conditioners in the EU. [38] Due to a gradual phase down of high GWP HFCs, the main trend is currently towards the mentioned low GWP natural refrigerants along with low GWP synthetic hydrofluoroolefins (HFO) and hydrochlorofluoroolefins (HCFO). A low GWP has a link to a short atmospheric lifetime of a substance. [17,22]

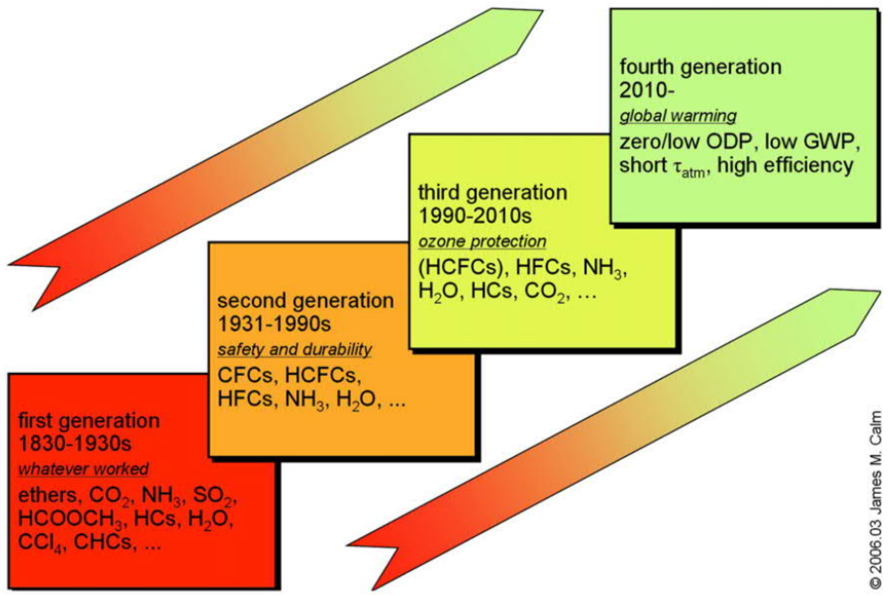

Figure 3.5. Refrigerant development [38].

The identified negative environmental impacts from the direct refrigerant emissions had caused a development of a regulatory framework during the refrigerant development [38]. These influencing the EU countries, up to today in 2020 and the future in the light of available data are summarized in the following.

The Montreal Protocol

The Montreal Protocol is a global agreement that was established in 1987 for protecting the stratospheric ozone layer by reducing the production and consumption of ODSs, finally phasing out ODSs. In developed countries, CFCs have been banned since 1996 in new systems, while since 2010 in remaining developing countries. On the global scale, HCFCs are being totally phased out by 2030 such that existing systems in 2020 can be serviced until 2030. The Montreal Protocol has been adjusted and amended several times. The most recent amendment (Kigali Amendment in 2016) included HFCs on the list of controlled substances indicating that the production and consumption of

HFCs should be gradually reduced in the world from a baseline that is defined on basis HFC consumption from years 2011…2013 expressed in CO_2-equivalent. The reductions should be implemented according to a phase down schedule that varies somewhat between different parts of the world. For the EU member countries, a first step included a reduction of 10 % from the baseline in 2019 and finally in 2036, a step includes a reduction of 85 % from the baseline. [39]

The F-gas Regulation

The F-gas Regulation is a legal requirement for a 79 % gradual reduction of HFCs in the EU market compared to a baseline that has been defined as the average CO_2-equivalent from experienced levels in 2009…2012. A reduction of 7 % from the baseline was allocated for 2016, while 37 % was for 2018 from the baseline. The next deadline is in 2021 when 55 % must be reduced and further after few deadlines, the 79 % must be reduced from the baseline by 2030. In addition to the phase down schedule, there are restrictions for the GWP in new systems that are specified by applications. HPs are not clearly mentioned in the application list, but e.g. the refrigerant GWP for new refrigerators and freezers for a commercial use will be limited to be less than 150 in 2022. [34]

3.3 Main components

This section aims to give a better insight from the main components of HPs. The main focus will be in the compressor because those that are suitable for HTHPs was one of the objectives. In the cycle configuration, there are several supplementary components that are necessary [22,23]. This thesis does not pay attention on those because the purpose was to focus in the main components.

3.3.1 Compressor

The compressor is a flow machine whose purpose is to circulate the refrigerant along with to compress the low pressure dry refrigerant vapor from the evaporator and raise its pressure to that of the condenser. The compressors can be divided into two types, positive displacement and dynamic from which turbo (270…36000 kW) compressor is the most common dynamic along with reciprocating (0.25…1000 kW), scroll (6.5…100 kW) and screw (75…6000 kW) compressors are the most common positive displacement. In the brackets of the previous clause, there were approximate cooling capacities that a single compressor can provide. [22] The heating capacities could be higher since the compression work extracted by the heat losses is supplied to a heat sink, too. In the following, the reciprocating, screw, scroll and turbo compressors will be shortly discussed.

A very widely used positive displacement type is the reciprocating (later, ‘’piston’’) compressor. The piston compressor is an adaptable in the refrigerant, size, number of cylinders, speed and method of drive. The piston compressor works on basis the 2-stroke cycle where an automatic pressure-controlled suction and discharge valves are used. As the piston descends in the suction stroke, the suction valve opens to suck the vapor from the evaporator. At the bottom of the stroke, the suction valve closes when the compression stroke begins. Further, the discharge valve opens when the cylinder pressure becomes higher than that in the discharge pipe and the compressed vapor flows to the condenser. The suction valve opens after the compression stroke when the cylinder pressure is lower than the suction pipe pressure. In practice, there is left some compressed vapor at top of the piston in the end of the compression stroke. This vapor must re-expand before a fresh vapor amount can be sucked. The larger amount of vapor remains in the cylinder in the end of the compression stroke, the higher are the compressor volumetric losses due to the re-expansion. [22]

A typical screw compressor has two rotors on parallel shafts next to each other. As these turns inside a closely fitting casing, the vapor between grooves of the rotors gets compressed. The screws compressors have a built-in volume ratio that causes that the pressure ratio is constant. The consumed electricity will be minimized only when the working pressure ratio corresponds to the built-in volume ratio [22,23]. The volumetric losses depend on the rotor tip speed, so smaller screws have to operate at higher speeds to maintain a good volumetric efficiency. An adequate lubrication and its maintenance are essential. The lubrication, cooling and sealing of the rotors is commonly assisted by an injection of oil along the rotors. The lubrication oil must be separated from the discharge vapor along with the oil is then cooled and filtered before being returned to the lubrication circuit. There are basically two types of screw compressors from which other has a one rotor and the other two rotors that are called as single and twin screw compressors respectively [22]. This thesis does not distinguish between these screw compressor types.

The scroll compressors compress the vapor with two inter-fitting spiral-shaped scrolls where the other scroll is stationary, and the second scroll moves in an orbit with a circular motion and compresses the vapor. The Scroll compressors have a built-in volume ratio like the screw compressors have. The scroll compressors have relatively low compression leakages resulting a good volumetric efficiency. The oil injection is not needed to the compression process, but the lubrication of bearings is vital. [22]

The most common dynamic compressor is the turbo (centrifugal). The turbo imparts energy to the vapor by increasing its velocity and then converts this to pressure. The vapor is sucked axially to the turbo rotor that has curved blades and discharges the vapor out tangentially. The compression pressure rises are not great, but the vapor can be compressed in one, two or more stages due to the impellers can be arranged next to each other. Only the main bearings require lubrication, thus turbo can run almost oil-free. There are also magnetic bearings that do not require the lubrication. [22]

Currently, the compressor sealing surfaces, and the bearings require the lubrication with the oil regardless of the compressor type. The oil gets is in touch with the refrigerant. With the screw compressors, the lubrication oil must be separated from the refrigerant after the compressor discharge. As for other compressor types, the oil separation can be needed depending on the case. [23]

The compression process generates heat. The compressor discharge temperature must be limited to avoid risk of the lubrication oil or the refrigerant decomposition. The discharge temperature depends on the refrigerant and operating conditions. Commonly, a sufficient oil cooling can be provided with the suction refrigerant if its temperature is not too high. A fan may be used for some extra cooling. However, if there is a risk that the discharge temperature can get excessive high, extra cooling with water can be needed. Extra oil coolers can be needed which can be water, air or refrigerant cooled. The compressor lifetime can decrease if the decomposition occurs. Thus, monitoring is necessary. The temperatures are typically monitored by using temperature sensors and an oil pressure sensor. The compressor is stopped if the temperatures rise too high or the oil pressure is too low. [22]

A HP system is designed to supply a maximum demanded heating capacity that is considered to be supplied by using the HP. However, for majority of the operation time, it can be that the HP operates at some lower heating capacities. A lower capacity requires capacity reduction devices. Currently, the most efficient way to control the capacity is to control the compressor speed by using a variable speed drive (VSD) to control the electrical motor speed that operates the compressor. [22,23]

The amount vapor compressed by a compressor will always be less than the physical compressor displacement. This implicates that larger compressor displacements are needed when compared to the actual compressor suction volume flow rate. All losses that influence the volume flow rate of a compressor can be expressed with the compressor volumetric efficiency that is a ratio of actual refrigerant volume flow rate at the compressor suction to the compressor displacement as shown in Equation 9. The losses include the piston re-expansion and other losses such as leakages. [22]

$$\eta_{vol} = \frac{\dot{V}_1}{\dot{V}_{comp,disp}} \tag{9}$$

The energy efficiency of a compression is defined as a ratio of the ideal isentropic compression to a real compression as shown in Equation 10. The enthalpies are numbered according to Figure 3.1 (like the refrigerant flow rate in Equation 9) where $h_2{}'$ is an actual enthalpy after the compression that is higher than the isentropic enthalpy h_2. The isentropic work input is the minimum required work to compress the vapor. The actual work will always exceed the isentropic work due to losses such as motor and pressure losses. The COP decreases with decreasing isentropic efficiency. [22]

$$\eta_{is} = \frac{h_2 - h_1}{h_2{}' - h_1} \tag{10}$$

In practice, the volumetric efficiency is approximately linearly, and the isentropic efficiency is generally decreasing with an increasing pressure ratio as approximately shown in Figure 3.6. On the other hand, the isentropic efficiency drops if the pressure ratio is too small. [22]

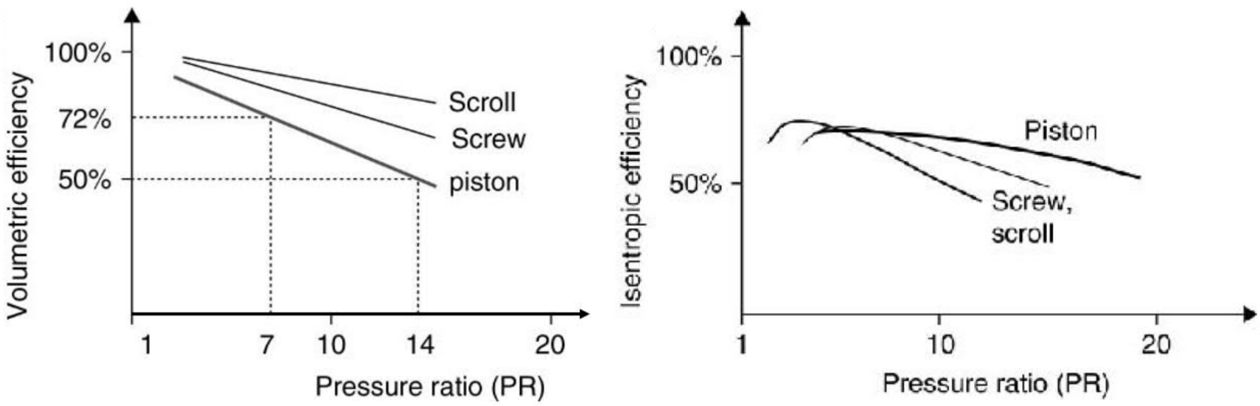

Figure 3.6. Approximate feature of the compressor volumetric and isentropic efficiencies [22].

3.3.2 Condenser and evaporator

The purpose of the condenser is to receive the superheated, high pressure refrigerant vapor from the compressor and to supply heat from the refrigerant to a heat sink by cooling first the vapor by removing the superheat and then the latent heat such that the refrigerant condenses to 100 % liquid along with to slightly subcool the liquid [22].

When neglecting heat losses along the cycle circuit and the compression, the total heat supply from the refrigerant to the heat sink is the sum of the heat absorbed with the evaporator from a heat source and the compressor electricity input. Separate desuperheaters can be used prior to the condenser to better utilize the temperature glide of the higher grade superheated vapor. The purpose of the evaporator is to receive the refrigerant in a low pressure and in a low temperature from the expansion valve along with to bring the refrigerant to a close contact with the heat source. The refrigerant evaporates by absorbing heat from the heat source until the refrigerant is 100 % vapor and is slightly superheated. [22] As for HPs, the refrigerant absorbs heat from a low-grade heat source, that is, the waste heat in this thesis.

As the condensers and the evaporators, air, water or other liquid are typically used to exchange heat with the refrigerant. Where the air is used, fans are typically used to force the air through a heat exchanger (HX) where the refrigerant circulates. Where the water or the liquid is used, e.g. shell-and-tube HXs or brazed plate HXs are used where the refrigerant and the water or the liquid circulate in separate channels. [22]

Heat transfer through the walls of the evaporator and the condenser HXs require a temperature difference between the refrigerant and the other fluid, e.g. the water. The larger HX heat transfer area, the lower will be the temperature difference at the pinch point (ΔT_{pinch}) where the temperature difference is the lowest. [22,40] The COP can be higher, being closer to the theoretical maximum when the HX area gets larger because T_{lift} gets lower, i.e. closer to the theoretical minimum. A larger evaporator HX area enables a higher T_{evap} that leads to a higher refrigerant density at the compressor suction meaning a higher VHC along with a lower pressure ratio and less compressor electricity consumption for a given heating capacity. A larger condenser HX area enables a lower T_{cond} that leads to a colder refrigerant liquid entering the expansion valve meaning a larger enthalpy difference in the evaporator that increases VHC along with a lower compressor discharge pressure meaning a lower pressure ratio and less compressor electricity consumption for a given heating capacity. [22] The influence of the evaporator and the condenser HX heat transfer area (UA is the thermal conductance of HX), i.e. ΔT_{pinch} is illustrated in Figure 3.7 where the potential improvement in terms of the COP is shown if ΔT_{pinch} is decreased from 10 °C (K) towards the theoretical 0 °C. If ΔT_{pinch} could be 0 °C, the practical COP could be as close the maximum achievable COP as possible in terms of the HX performance. [16]

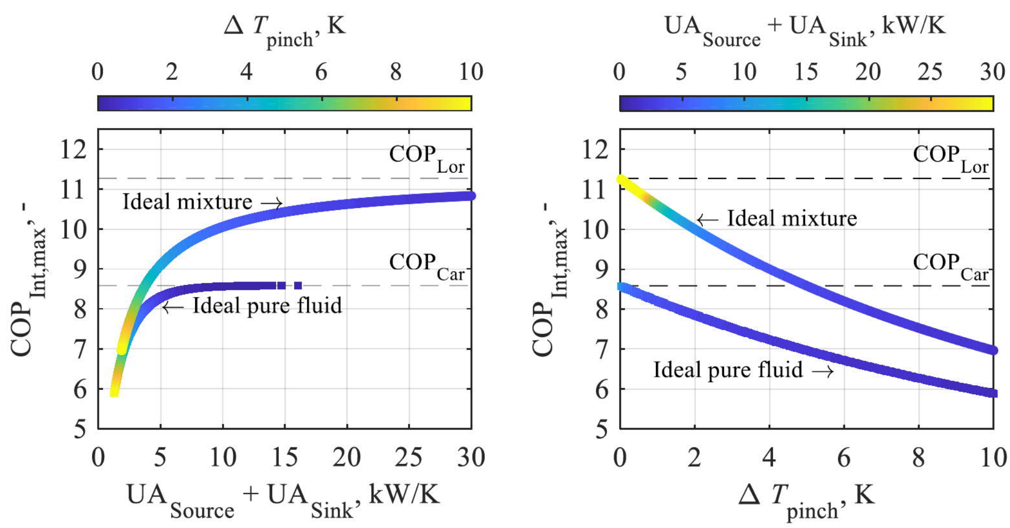

Figure 3.7. Influence of the evaporator and condenser HX area on COP [16].

3.3.3 Expansion valve

The purpose of the expansion valve is to control the refrigerant flow from the high pressure condensation side to the low pressure evaporation side. The expansion valve must ensure that there is no risk of liquid refrigerant flowing to the compressor. Thus, the expansion valve is controlled such that the refrigerant superheats in the evaporator, typically of the order of 5 °C. [22] A thermostatic expansion valve (TEV) and an electronic expansion valve (EEV) are commonly used from which the conventional TEV is typically used in small HPs and the EEV in larger and industrial HPs [23]. These are described shortly in the following.

The TEV consist of the actual valve located right before the evaporator, a sensor element that is connected to the compressor suction pipe right after the evaporator and a capillary tube that is between the valve and the sensor. The sensor and the capillary tube form a separate system that is filled with the same refrigerant that is in a HP system. The main component inside the valve is a spring that controls an orifice via a pin. The saturated vapor refrigerant has a lower saturation pressure in the evaporation temperature when compared to the temperature after the required superheat. The refrigerant is theoretically at a constant pressure in the whole evaporation side of the HP along with the refrigerant stays as a saturated vapor in the sensor system according to the HP system refrigerant temperature after the evaporator. Thus, the sensor system refrigerant pressure rises if the superheat rises or its pressure decreases if the superheat decreases. The spring is pre-set to correspond the refrigerant pressure difference between the HP and sensor systems, i.e. the desired superheat. If the thermal power from the heat source $\dot{Q}_{source}$ increases, the superheat could increase without the thermostatic control of the TEV. Thus, the pressure increases in the sensor system and this pressure pushes the pin to open the orifice more to increase the HP system refrigerant flow to the evaporator and further, the superheat decreases due to increase in the flow rate. If the thermal power from the heat source $\dot{Q}_{source}$ decreases, the operation is vice versa. [22]

The EEV offers a finer superheat control, i.e. the compressor protection. The higher superheat in the evaporator, the lower evaporation temperature is needed to satisfy the ΔT_{pinch} in the evaporator if the HX area of the evaporator is fixed. The finer superheat control allows a smaller superheating in the evaporator without a risk of the refrigerant liquid passing to the compressor and thus, allows a higher evaporation temperature that leads to a higher COP due to a lower T_{lift}. In each case, the EEV consists of the valve and an electronic controller. The orifice opening is actuated with a motor and the controller controls the motor according to a pressure sensor and a temperature sensor that are located right after the evaporator outlet. The controller is pre-configured according to the refrigerant properties and the valve type. The superheat can be controlled with this setup. [22]

4 Commercialized high temperature heat pumps

This section summarizes current commercialized vapor compression HTHPs and gives an overview from their key features. The considered features are the applied refrigerant, the maximum achievable heat sink temperature, the heating capacity and the compressor technology. Table 4.1 shows the current commercialized HTHP models that can produce at least 85 °C heat sinks. All in all, 32 HTHP models were found from 16 manufacturers. The most common refrigerants are R134a, R245fa, R717 and R1234ze(E). The maximum achievable heat sink temperature is 165 °C, while several models can achieve at least 100 °C heat sinks. The heating capacities vary considerably across the models being from around 10 to 20000 kW. The piston compressor and the screw compressor seem to be the most common HTHP compressors while the turbo compressor is applied in few models.

Natural refrigerants are applied in several models from which R717 and R744 are the most common. From models that apply synthetic refrigerants, high GWP HFC refrigerants R134a and R245fa seems to be the most common. However, several models apply a low GWP synthetic HFO refrigerant R1234ze(E). In addition, a model HBS4 applies several different synthetic low GWP HCFO and HFO refrigerants. The maximum heat sink temperatures give a feeling from the maximum practically achievable temperature for a certain refrigerant with the current technology. For example, it seems that the heat sink temperature for R717 is limited to be around 90…95 °C, whereas R1234ze(E) could achieve 95…100 °C heat sinks. When looking at the heating capacities and the compressor technologies, it can be observed that the piston compressors are typically used for smaller heating capacities, the screw compressors from small to large heating capacities and the turbo compressors for large heating capacities.

Table 4.1. Commercialized HTHPs currently (extended and updated from [17]).

manufacturer	product	refrigerant	$T_{sink,max}$ [°C]	Q_{min} [kW]	Q_{max} [kW]	compressor	reference
Kobe Steel	SGH165	R134a/R245fa	165	70	660	screw	[40,41]
	SGH120	R245fa	120	70	370	screw	[40,41]
	HEM-HR90	R134a/R245fa	90	70	230	screw	[41,42]
Viking Heat Engines	HBS4	R1336mzz(Z)	165	50	250	piston	[43,44]
		R245fa,R1234ze(E),R1233zd(E),R1336mzz(E)					
Ochsner	IWWDSS R2R3b	R134a/R245fa	130	170	750	screw	[45]
	IWWDS ER3b	R245fa	130	170	750	screw	[45]
	IWWHS ER3b	R245fa	95	60	850	screw	[45]
	IWWHS ER6c2	R1234ze(E)	90	60	850	screw	[45]
Frigopol	High Butane	R600	130	10	50	piston	[27,46]
Hybrid Energy	Hybrid Heat Pump	R717/R718	120	250	2500	piston	[47,48]
Mayekawa	Eco Sirocco	R744	120	65	89	piston	[49,50]
	Plus+HEAT NHS *	R717	95	487	1218	piston	[51,52]
	HeatCO2m	R744	90	45	110	piston	[52]
	Plus+HEAT NHK	R717	85	244	487	piston	[51,52]
Combitherm	HWW R245fa	R245fa	120	62	252	piston	[53]
	HWW R1234ze	R1234ze(E)	95	85	1301	screw	[53]
Oilon	ChillHeat P	R1234ze(E)	100	30	450	piston	[54]
	ChillHeat S	R1234ze(E)	85	180	2000	screw	[54]
Friotherm	Unitop 22	R1234ze(E)	95	600	3600	turbo	[55,56]
	Unitop 50	R134a	90	9000	20000	turbo	[55]
Star Refrigeration	Neatpump	R717	90	350	8000	screw	[57]
GEA Refrigeration	Custom 63 bar	R717	90	300	15000	screw	[58]
Johnson Controls	HeatPAC	R717	90	341	1346	piston	[59]
	DualPAC	R717	90	460	1841	piston	[59]
	NS heat pump	R717	90	3260	7090	screw	[59]
	Titan OM HP	R134a	90	5000	20000	turbo	[60]
Mitsubishi	ETW-L	R134a	90	376	547	turbo	[40,61]
Viessmann	Vitocal 350-HT Pro	R1234ze(E)	90	62	428	piston	[62]
Advansor	VALUEPACK	R744	90	30	96	piston	[63,64]
	STEELXL	R744	90	300	1400	piston	[63,64]
Trane	RTWF G	R1234ze(E)	85	363	1470	screw	[65]

* potentially available soon

5 Potential applications

Integration of HTHPs in an industrial site should satisfy following conditions. Firstly, large quantities of waste heat that cannot be utilized with direct HXs should be available. Secondly, the temperature levels of the waste heat and the heat demand along with their quantity should be matched such that HTHP integration is possible. Thirdly, these heat flows should be as simultaneous as possible. [66] The waste heat should be utilized as near as possible to its origin since the investment costs are typically higher when transferring longer distances thermal energy. According to this principle, the waste heat utilization should be prioritized from the top to bottom like listed below. [5]

- Improve energy utilization to reduce the energy consumption and the waste heat generation;
- Recycle the waste heat back to the same process where it is generated or if not possible, to other processes that are as near as possible;
- Recycle the waste heat to another industrial actor within the same industrial area;
- Recycle the waste heat to a district heating network.

During the literature survey, it was noticed that temperature levels for industrial waste heat and heat demand are similar across several countries. In the light of this, it was assumed that the temperature levels could be similar in other countries, too. Also, it was assumed that the temperature distribution of the total heat demand in industrial sectors is similar across countries. In section 5.1, Finnish energy potentials along with the temperature levels are illustrated. Further, promising sectors and processes are discussed in sections 5.2 and 5.3 respectively.

5.1 Energy potential

This section provides data from the Finnish energy consumption and from an estimated technical Finnish waste heat energy potential along with from the temperature levels that were found from the literature. There was no data available from the Finnish waste heat energy potentials concerning textile, transport equipment, plastic and machinery industries. In addition, no data was available from the transport equipment sector waste heat temperature levels. Appendix 2 shows the data in a tabular form and their references.

The technical waste heat potential is an energy quantity that is possible to recycle with existing technologies. A major proportion of the Finnish technical waste heat potential is below 55 °C, and even below 100 °C. However, all the technical potential is not necessarily recyclable in an economic way. [5]

Figure 5.1 shows the total energy consumption (demand) from 2018 and the estimated waste heat potential per industrial sector along with Figure 5.2 shows the temperature levels per industrial sector. The waste heat energy quantities were extrapolated from year 2010 to the present by assuming that the waste heat quantity varies in the same way as the total energy consumption as assumed in [5]. The energy potentials are the largest in paper and pulp, metal refining, chemical and oil refining industries. The temperature levels show large variation for waste and demand. The temperatures were listed up to 1000 °C. As shown in Figure 5.2, the transport equipment industry waste heat temperature levels could be similar with machinery industry since the heat demand is at similar temperatures. The industrial energy potentials and the temperature levels show that there can be applications for HTHPs in every industry since both the waste heat and the heat demand can be below or near 100 °C.

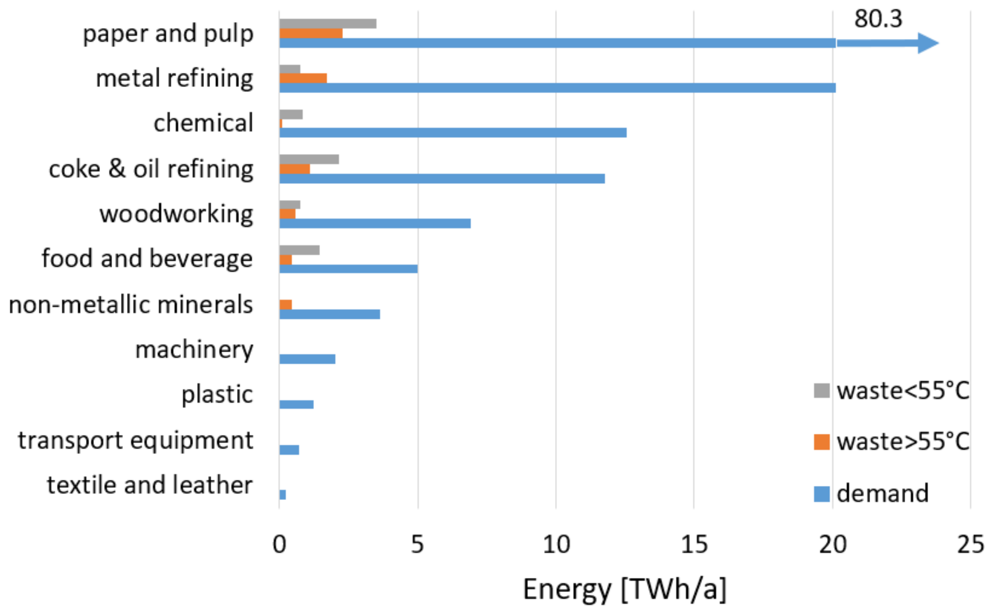

Figure 5.1. Energy potentials in the Finnish industry, ''demand'' = total energy consumption, ''waste<55°C'' = waste heat below 55 °C, ''waste>55°C'' = waste heat above 55 °C.

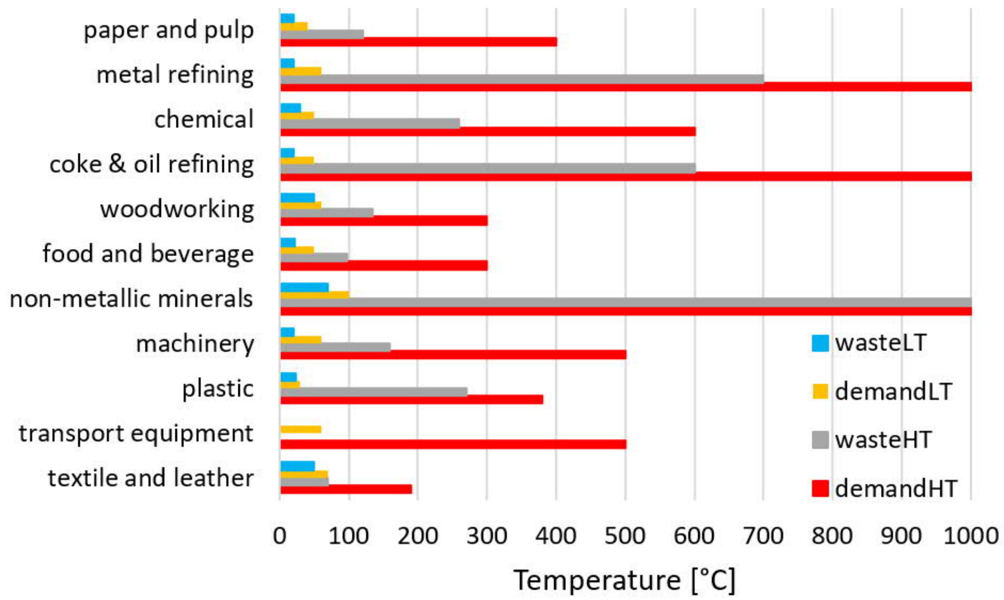

Figure 5.2. Temperature levels in industry, ''wasteLT'' = waste lower T limit, ''wasteHT'' = waste higher T limit, ''demandLT'' = demand lower T limit, ''demandHT'' = demand higher T limit.

In addition to the industrial potentials, Figure 5.3 shows DH energy potentials and temperature levels. The DH consumption (demand) is the experienced from year 2019 and is the outdoor air temperature fixed. From the energy bars in Figure 5.3, two uppermost bars show an estimated HTHP capacity potential and an existing HTHP capacity in the Finnish DH networks respectively. Two bottommost bars show the waste heat potentials that were found from the literature. The lower part of Figure 5.3 shows the temperature levels of the waste heat sources that are considered suitable for HTHP connected to a DH network. In the current Finnish DH networks, the supply water (demand) dimensioning temperature is 120 °C and is typically between 75…115 °C depending on the outdoor air temperature [67]. The lower and higher temperature limits for the waste and the demand show a smaller deviation when compared to the industry. In case of flue gases, the temperature is considered to be the evaporation temperature along with the combined cooling heating and power (CCHP) for a waste heat from absorption chillers. The DH network lower waste heat temperature is considered to be from a district cooling return pipe and the higher waste heat from a DH return pipe.

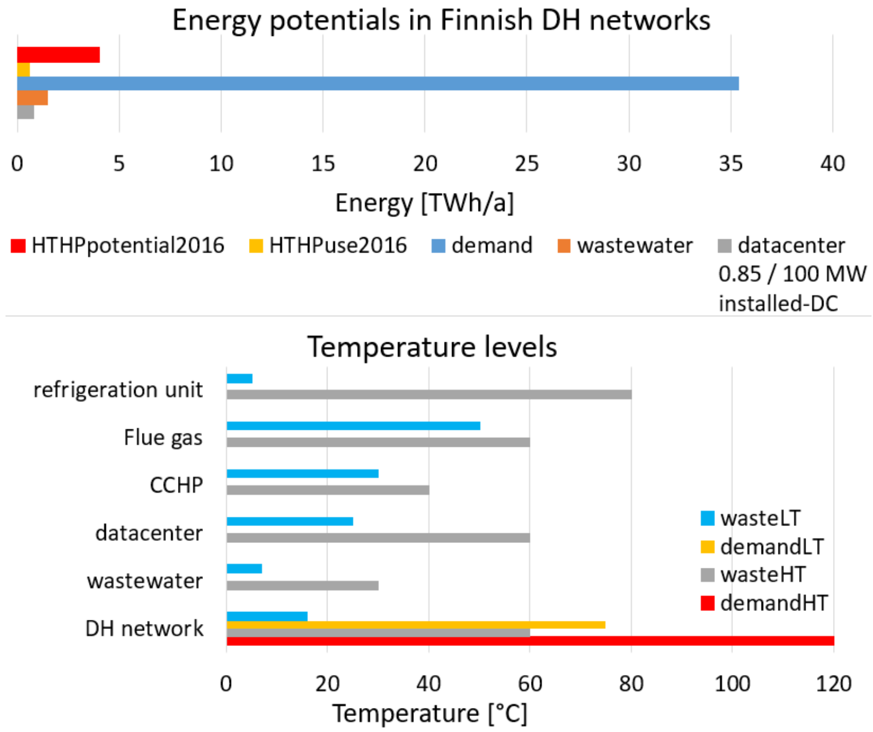

Figure 5.3. Energy potentials and temperature levels in Finnish DH networks.

The demanded DH and the potential waste heat temperature levels show that there is an evident technical potential for HTHPs in the DH networks. Even for the minimum waste heat temperature from refrigeration units and the maximum DH supply temperature, T_{lift} could be around 115 °C. This relatively high T_{lift} could be technically possible [50]. The DH supply water temperature is below 100 °C for outdoor air temperatures down to around -15 °C [68]. According to the test reference year 2012 for the southern Finland [69], the annual outdoor air temperature duration for temperatures lower than -15 °C is around 155 hours. This indicates that HTHPs for heat sinks up to 100 °C could achieve the demanded DH supply temperature for a major proportion of a year. As for this DH supply temperature of 100 °C, T_{lift} could be sufficiently low with several heat sources to achieve a good practical COP [17]. Note that waste heat from the industry can be recycled into a DH network or, heat can be recycled from the sources listed in touch with the DH potentials to the industry. However, the earlier mentioned industrial waste heat prioritization principles should be kept in the mind.

5.2 Sectors

This section discusses sectors that have been noticed to have promising applications and energy potential according to the literature. In addition, these discussed sectors are promising with respect to the energy potentials and the temperature levels that were illustrated in the previous section. This agreement gives evidence that these sectors can be promising e.g. in Finland or in some other country, too.

Paper and pulp industry

In the EU, USA and in several separate EU countries, the paper and pulp industry has a relatively large heat demand below 150 °C. Several potential processes have been identified for heat sinks below 150 °C. [17,18,20,40,48,50,66,70,71] Promising processes have been identified to be e.g. paper and wood drying along with evaporation processes [72]. The most effective improvement potential has been expected to be the waste heat recovery while the paper and pulp processes are not expected to change significantly [12]. Challenging for HTHP is that e.g. the Finnish paper and pulp industry generates inexpensively a major part from the energy demand from a by-product biomass [72].

Food and beverage industry

In the EU, USA and in several separate EU countries, the food and beverage industry has relatively large heat demand below 150 °C. Several potential processes have been identified for heat sinks below 150 °C. [17,18,20,30,40,48,50,66,70,71] The most promising processes have been identified to be the drying and the washing (recycle latent heat from exhaust air), the pasteurization (simultaneous cooling and heating) along with a general process heating and cleaning with a hot water [17,20,73].

Chemical industry

In the EU, USA and in several separate EU countries, the chemical industry has relatively large heat demand below 150 °C. Several potential processes have been identified for heat sinks below 150 °C. [17,18,20,40,48,50,66,71] Especially, boiling processes have been identified to be promising [17,20].

Woodworking industry

In the EU, the woodworking industry has lots of heat demand below 150 °C. Several potential processes have been identified for heat sinks below 150 °C. [17,20,40,50,66,71] Especially, the drying processes have been identified to be promising [17,20].

District Heating (DH) & District Heating and Cooling (DHC) networks

This is a one evident application for HTHPs [74]. HTHPs can become more attractive in Finland if the government's recent proposal to decrease the electricity taxation for HPs that generate heat to DH networks becomes valid along with the EU directive 2018/2001 regulates to use the waste heat in DH [5]. The climate targets and a potential price increase of emission allowances forces energy companies to develop more energy efficient heat supply. Increasing VRE generation can potentially fluctuate the electricity price and decrease the electricity price if excess VRE generation occurs. HTHP potential has been noticed to increase if the electricity price decreases. Thus, usage of HTHPs in recycling the waste heat has been expected to grow in the light of a more VRE based electricity grid in the future where the excess VRE could be balanced e.g. by using HTHPs. HTHPs can enable a bidirectional DH network where the customers could both buy and sell heat. Both waste heat producers and DH network operators could obtain economic benefits from the heat recycling. [75-77] The heat and electricity grids should be integrated by using HTHPs to enable heat supply without combustion processes [78]. For instance, a Finnish energy company HELEN utilizes HTHPs for the DH or for simultaneous cooling and heating where the heat source is depending on the situation wastewater or cooling the return water of the district cooling network [8,74]. The waste heat recycling can be turned into a business for which an example is that HELEN buys heat from customers according to their tariff and offers contracts where the DH supply is based on a recycled waste heat with HTHPs [79]. To add, there were 66 HTHPs connected to Danish DH networks in 2019 [80].

In general

A technical industrial heat supply potential with HTHPs could even double and number of potential industrial processes could significantly increase if the maximum heat sink temperature increases from 100 to 150 °C [17,20,66,70,71]. For instance, as for Germany in 2010, it has been evaluated that if the maximum HTHP heat supply temperature increases from 80 to 140 °C, the technically possible heat supply with HTHPs could increase from 14 to 32 % of the total industrial heat demand [20].

5.3 Processes

Promising industrial and general processes are discussed in this section. Appendix 3 gives a comprehensive overview from industrial processes that could be suitable for HTHPs according to the demanded $T_{sink,out}$. Especially, the required temperature levels, the efficiency improvement potential and links to the promising sectors determined the promising processes.

Simultaneous heating and cooling

The waste heat source cools when utilizing it with the evaporator. The profitability of HTHPs can increase significantly if this cooled waste heat can be utilized in a cooling process. [74] HTHP can simultaneously provide cooling by absorbing the waste heat with the evaporator and provide heating by supplying the heat with the condenser. Separate heating boilers and water chillers are conventionally used, thus HTHP could even replace both those. The COP can be exceptionally high in these situations. [8,73] The improvement on the COP for simultaneous cooling and heating can be observed from Equation 2. An example application is the pasteurization [73].

Industrial drying and washing

The drying is one of the world's most widely used industrial process and often occurs at temperatures near or below 100 °C [20]. A conventional dryer type circulates hot air over moist products from which the moisture evaporates into the air and decreases the air temperature. A proportion of this cool and moist air is exhausted to outdoors and a proportion is recirculated and mixed with outdoor air back to dryer inlet via a heating coil. In this drying process, a major part of the exhaust air energy content is within the moisture as a latent heat. This latent heat cannot be recycled with direct heat exchangers due to the exhaust air temperature is too low. However, HTHP can recycle the latent heat by condensing the moisture with the evaporator and by releasing it with the condenser at a required higher temperature. HTHP can recover heat up to 80 %, while the rest remains in dried products and some heat is lost to the environment. Similar domestic HP cloth dryers are widely used currently. Similar latent heat recycling has potential also in the industrial product washing. [20,48,72,73,81] It is estimated that 12…25 % of a national industrial energy consumption in developed countries is caused by the industrial drying where majority of the energy is supplied by using fossil fuels [81]. Thus, there is probably a large application potential for HTHPs across the world. More evidence on the drying potential gives that this process is needed in several industries as shown in Appendix 3.

Flue gas condensation and wet flue gas scrubbers

The principle corresponds the drying because moisture latent heat is recovered from the flue gases by condensing the moisture. HTHP can improve the latent heat recycling due to direct heat exchangers cannot capture the latent heat when the flue gases are cooled enough. The flue gases originate generally from heat-only or CHP boilers that generate process heat or provide heat to a DH network. The recovered heat can be recycled e.g. to the DH network or to industrial processes to reduce the fuel use and thus, to reduce the emissions and the operational costs. [18,50] HTHP can be used in conjunction with wet flue gas scrubbers to improve the heat recovery from the flue gases. The improvement is principally due to higher amount condensing of the flue gas moisture with a usage of the HTHP. By adding a HTHP to the process, the heat recovery can be even 2…3 times higher when compared to a basic flue gas scrubber without the HTHP [82].

Datacenters (DC)

The DCs require lots of cooling. Depending on the cooling solution, the waste heat can be captured at a temperature between 25…60 °C. This waste heat is a potential waste heat source for HTHPs to be recycled to DH networks and even, to the industry. Even up to 97 % of the consumed electricity could be captured as the waste heat. DCs are estimated to account for up to 1.5 % of the world's total electricity consumption already in 2010, while DC capacity has been rapidly growing recently. Nordic countries, such as Finland, are especially suitable for DCs due to free cooling possibilities. There are several existing DCs e.g. in Finland from which the waste heat is recycled to a DH network. [83-84] The number of DCs have been growing during recent years and more is being designed, thus DCs are expected to be increasingly important heat sources, especially in the DH production [5].

Thermal Energy Storages (TES) and electricity grid stabilization

It can be that the waste heat generation and the heat demand are not simultaneous. Storing of a low temperature waste heat for a later use might be necessary to enable a stable heat source condition for HTHP that is a precondition for a stable HTHP operation. [27,85] As mentioned earlier, the VRE generation is expected to increase in the future. This indicates that there is an increasing need to stabilize the electricity grid when the VRE generation is too high. When the VRE generation is too high (and electricity price is low), the waste heat can be with a low cost and without environmental impact stored at a high temperature to TES by using HTHP. Further, the high temperature TES can be discharged when the heat is demanded and simultaneously, the VRE generation can be possibly low indicating higher electricity prices. [86,50,72] As an additional example, HTHPs can be used to store the VRE to TES when the VRE generation is too high. Then, TES can be discharged when the VRE generation is low to generate electricity with a heat engine. [50]

Combined Heating and Power (CHP) & Combined Cooling Heating and Power (CCHP)

HTHPs can aid to balance the electricity grid in high VRE generation times, while the CHP plants could act as a back-up capacity. Electricity can be produced to the grid with CHP when the electricity price is high, whereas the electricity can be bought from the grid when the electricity price is low and generate demanded heat by using HTHPs. Acting like this, industrial actors can contribute in balancing the supply and the demand in the electricity grid. [12] There are many ways to integrate HTHPs to the CHP production. As an example, HTHPs can be used to increase the heat production without increasing the electricity production. [87] In addition, HTHPs show an efficiency improvement and a lifecycle-cost saving potential in CCHP production plants [88].

6 Technical special solutions required for higher temperatures

HPs in the building sector intended for space and domestic hot water heating can typically supply heat sinks up to 70 °C. Because of the higher temperatures in touch with HTHPs, the technological requirements for the system components are higher along with the system design has important consequences on the system performance. [12] As mentioned in section 1.1, the heat exchangers and the expansion valves were not surveyed since the client considered that these do not differ from those that are commonly used in HP systems. But then, the refrigerant and the compressor were considered to be important. This is confirmed e.g. in [12,17,89]. In general, the surveyed literature did not show special considerations that should be considered in touch with the heat exchangers and the expansion valves when supplying higher temperatures.

The refrigerant is one of the most crucial, maybe the most crucial, in touch with HTHPs. The first step in the development and design of HTHPs is to find a suitable refrigerant with a good performance. There are various important requirements for the refrigerant. [12,17,21,89,90] The refrigerant and the compressor seem to be clearly the most stringent limiting factors for the operating domain of HTHPs. The compressor has technological restrictions related to the temperature and the pressure, whereas the refrigerant is limited by its critical point. The technological development can handle the issues concerning the compressor, whereas the critical point is a natural property that can be handled with an appropriate refrigerant selection. [91,92] Thus, important considerations concerning the refrigerant and the compressor were surveyed in detail and is the subject of this section. Only the subcritical cycles are considered unless else mentioned.

6.1 Refrigerants

When comparing the refrigerants, their properties vary significantly. If the cycle conditions such as the cycle configuration and T_{lift} stays fixed, the current research provides such signals that the refrigerant selection has the most significant influence on the HTHP performance. [12,17,25,89,92] This section discusses important refrigerant properties and how those influence on the cycle performance along with how these properties can be compared across the refrigerants.

6.1.1 Ideal properties

The thermodynamic properties of most of the refrigerants used in refrigeration units and in a low temperature heating are not suitable for HTHPs. Those refrigerants are typically restricted by a too low critical temperature leading to that the compressor technology is restricted because those refrigerants could have too high vapor pressures in the higher temperatures. The synthetic high GWP HFC refrigerants R134a and R245fa along with their mixture have been used for the HTHP development (Table 4.1). However, high GWP refrigerants are being phased down in the light of the current regulations. Thus, only natural and synthetic refrigerants that are environmentally friendly should be considered. [17,21,28,89,93,94]

Some of the most general properties for an ideal refrigerant will not be repeated here (see section 3.2.1). In the following, this section discusses the properties that are relevant for HTHP refrigerants. The properties are discussed according to [12,17,21,24,25,29,89,93,95] unless else cited. Note that hardly any refrigerant can satisfy all those selection criteria with the most favorable properties. Different operating conditions can require different refrigerants. Thus, the purpose is to find the most suitable refrigerant that performs well within a specific operating range. [21]

Ozone depleting and global warming potentials

First of all, ODSs are banned, i.e. ODP must be practically zero. In addition, GWP should be enough low in the light of the legislation. Preferably, GWP should be as low as possible (less than 10) in the light of the climate targets and to ensure that the refrigerant is future-proof.

Flammability

The refrigerant flammability is a significant safety issue and should be avoided if possible. If the refrigerant is flammable, significant and expensive safety measures are mandatory. According to SFS-EN 378-1:2016 [36], refrigerants are classified as non-flammable (1), mildly flammable (2L), flammable (2) and highly flammable (3). Refrigerants with low or no flammability (1, 2L, 2) reduces the complexity of the safety measures and thus the costs.

Toxicity

The toxicity is a significant safety issue as well. Non-toxic refrigerants are preferable to avoid hazards to persons. Certain precautions must be implemented as for toxic refrigerants. According to SFS-EN 378-1:2016 [36], refrigerants are classified as non-toxic (A) or toxic (B) along with toxic refrigerants have charge restrictions depending on the system location.

Coefficient of performance

The most important performance property is a theoretically achievable COP, that is more accurately, the COP that could be achievable in the practice. The higher COP, the better.

Volumetric heating capacity

The VHC is an important performance property influencing the compressor size and thus the investment costs along with influences the practically achievable COP. The higher VHC, the better.

Normal boiling point

The normal boiling point (NBP) should be a low to assist in achieving a high VHC. In addition, the NBP should be lower than the ambient air temperature to keep the whole HTHP system over-pressurized to avoid risk of the ambient air leakage to the system. A relatively high NBP restricts the HTHP use within lower waste heat temperatures.

Critical temperature

The critical temperature T_{crit} should be clearly higher than the condensation temperature T_{cond} to enable efficient subcritical cycles. T_{crit} must be at least 10…15 °C higher than that of T_{cond} to achieve a reasonably high COP. As for the subcritical cycles, the closer T_{cond} is to T_{crit}, the smaller is the condensation enthalpy and thus, negatively influences the COP.

Critical pressure

The critical pressure p_{crit} should be enough low, preferably less than 30 bar because it influences the system pressure level that is an important for safety aspects and equipment material loads. However, the pressure should be over 1 atm to avoid the air leaks.

Pressure ratio

The pressure ratio p_{ratio} is refrigerant dependent for a given operating condition and should be as low as possible to maximize the compressor volumetric and isentropic efficiencies, but not too low to avoid the drop in the isentropic efficiency (Figure 3.6). Thus, increasing pressure ratio decreases the COP and leads to that a larger compressor could be required for a given heating capacity.

Molar mass

In general, an enough high molar mass M causes that a special superheating is necessary to prevent the compression from ending to the two-phase area. The higher molar mass, the higher could be the demand to superheat the compressor suction refrigerant. [25]

Thermal stability and lubrication oil

The refrigerant and lubrication oil must be thermally stable in the desired operating range and the oil must maintain good lubricating properties at high temperatures.

6.1.2 Performance

When comparing simultaneously both the theoretical COP and the VHC across refrigerants, a good estimation of the performance for finding the most suitable refrigerant is possible. The COP is a direct performance indicator for a relative electricity consumption influencing the operating costs and potential emissions from the electricity consumption depending on the energy mix of the electricity generation. On the other hand, the VHC is an indirect performance indicator for the investment costs and influences the practical COP. In general, the COP and the VHC tends to be such that one or other is high and the other is low, thus a compromise must be done. [17,21,24,25,29,88,89,96-98] This section discusses the major performance indicators COP and VHC along with how those vary with respect to the refrigerant properties. The discussions in this section are based on [17,50,24,25,90,99] unless else cited.

Figure 6.1 exemplary shows the vapor pressures (left) and theoretically simulated COP with a constant T_{lift} of 50 °C (right) for several synthetic refrigerants that are suitable for the HTHP temperature range. The critical points are illustrated with dots. As shown, the higher refrigerant T_{crit}, the higher is the maximum theoretically achievable COP when T_{lift} is constant and T_{cond} varies. This behavior is linked to that the energy quality of the heat source increases with increasing heat source inlet temperature $T_{source,in}$. The increasing COP behavior can be confirmed by using Equation 3 and Equation 4 when considering a constant T_{lift} and by increasing T_{cond}. In addition, the subcritical operation is possible in an efficient way with the higher temperatures when T_{crit} is higher because the condensation enthalpy is not too low when there is a sufficient temperature difference between T_{crit} and T_{cond}.

As shown in Figure 6.1, the refrigerant dependent COP is maximized with T_{cond} of around 40…50 °C below T_{crit}. Depending on the cycle configuration and the operating conditions, the COP is maximized with T_{cond} of 20…60 °C below T_{crit} [17,50,98-100]. Thus, the mentioned minimum 10…15 °C temperature difference between T_{crit} and T_{cond} causes already some reduction in the COP as shown in Figure 6.1. As the optimal T_{crit} depends on the cycle configuration and operating conditions, the refrigerant dependent COP for heat sinks e.g. of 100 °C is optimal when T_{crit} is roughly between 120…160 °C. As for the vapor pressure, it can be noticed from Figure 6.1 that a higher vapor pressure can slightly influence the COP in a positive way. For instance, when comparing R1233zd(E) and R1336mzz(Z), R1233zd(E) has a lower T_{crit} but a higher p_{crit} and has a higher COP. In addition, the vapor pressure varies significantly across the refrigerants at some fixed temperature indicating varying pressure level requirements for the system and the compressors.

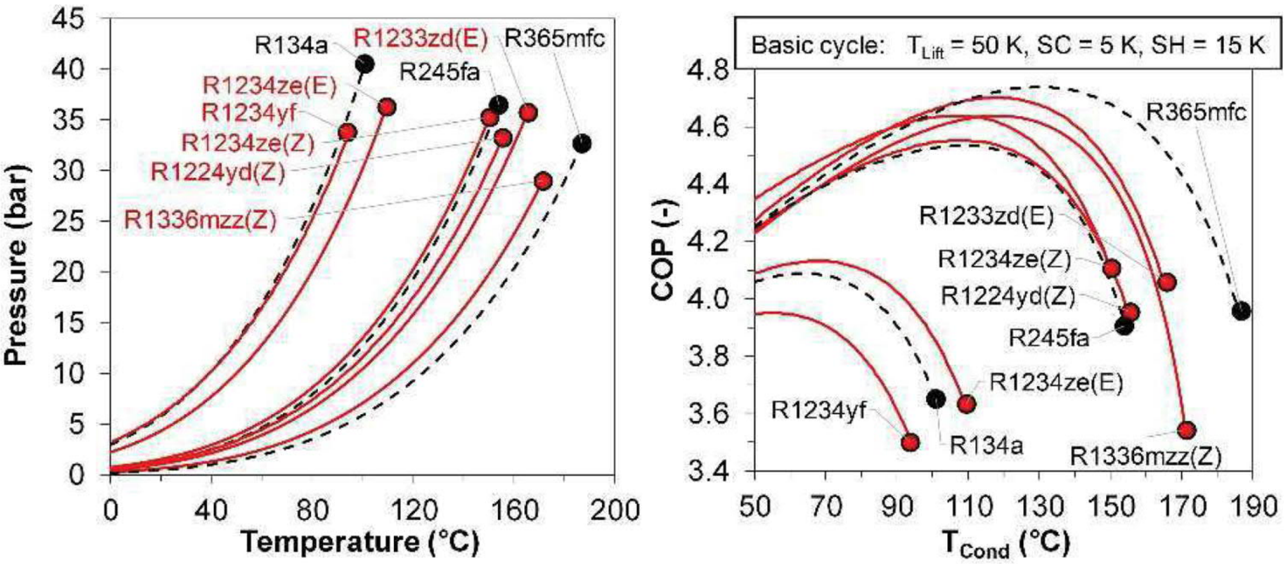

Figure 6.1. Dependence of COP on the critical temperature and the condensation temperature [98].

Figure 6.2 exemplary shows the dependence of a theoretically simulated VHC on the vapor pressure and T_{cond} with a constant T_{lift} of 70 °C as for several refrigerants that are suitable for the HTHP temperature range. The critical points are again illustrated with the dots. Like illustrated with the green dotted lines, the VHC is equalized across the refrigerants when the vapor pressures are equalized. So, the VHC increases when the vapor pressure increases. This suggests that the vapor pressure should be as high as possible to maximize the VHC. One refrigerant that is not shown in Figure 6.2, is a high pressure refrigerant R717 (ammonia) whose VHC is exceptionally high, e.g. at T_{cond} of around 80…100 °C being even around 6 times higher than that of R1234ze(Z) when the operating conditions are the same with both of the refrigerants [88,89].

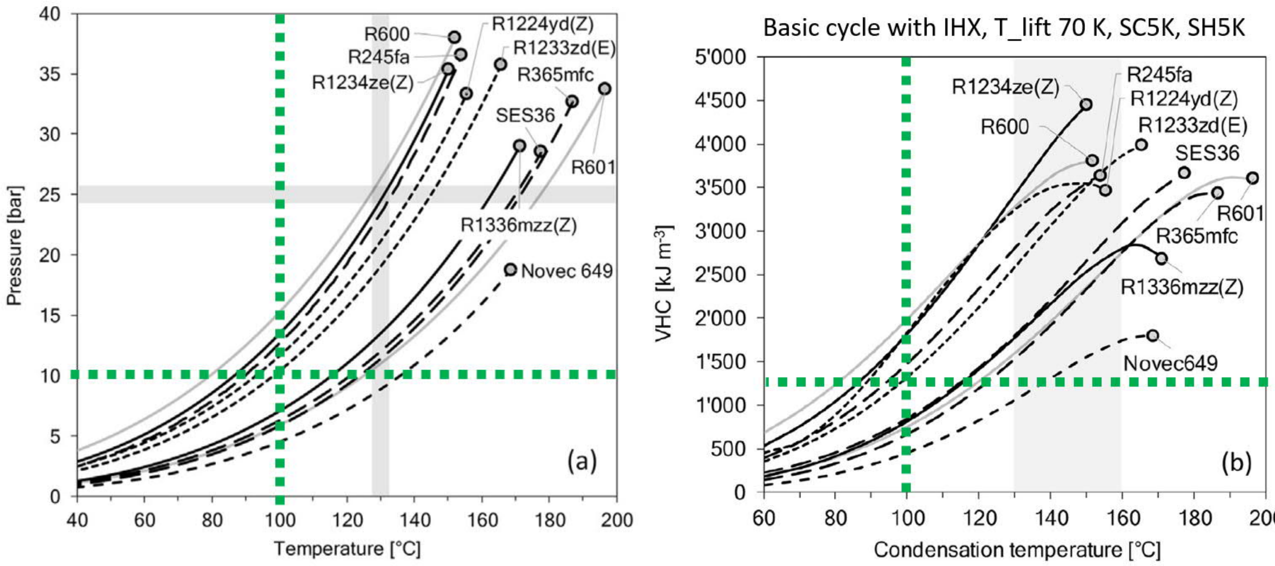

Figure 6.2. Dependence of VHC on the vapor pressure and the condensation temperature [17].

A too small VHC negatively influences the COP that can be achieved in the practice. It has been found that the VHC shall not be lower than 1000…2000 kJ/m^3 [17,19,25,29]. Figure 6.3 shows how an experimentally measured compressor electrical power per processed refrigerant mass flow rate is dependent on the VHC. This indicates that a lower practical limit for the VHC could be around 1500 kJ/m^3 to be sure that the VHC does not has a major negative influence on the practical COP. As the compressor electricity consumption increases per processed mass flow rate, the COP decreases. [25]

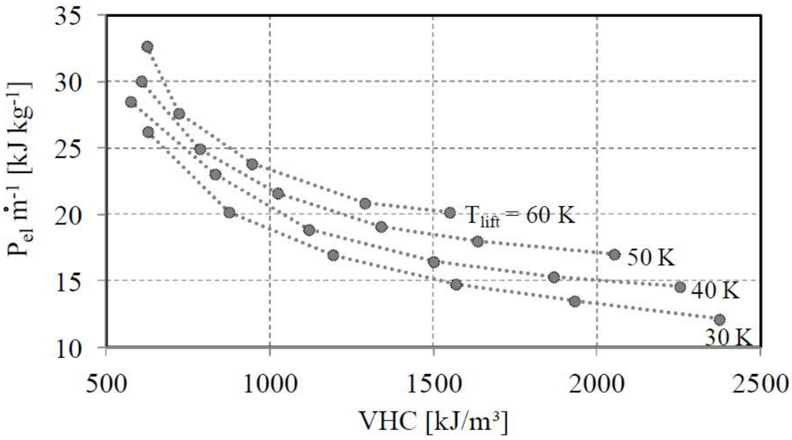

Figure 6.3. Dependence of a relative compressor electricity consumption on VHC [25].

6.1.3 Minimum required superheating

Different refrigerants vary significantly in terms of how the temperature develops during the compression [12,17,25,37,93]. The T-s-diagram is often used to determine the refrigerant behavior during the compression like exemplary shown for three refrigerants in Figure 6.4, where the two-phase regions are bounded with the saturated liquid and vapor curves like in the log(p)-h-diagram. The compression is the vertical line that is assumed to be isentropic, i.e. the entropy is a constant.

Three different saturated vapor curves can be distinguished that are often called as (a) bell-shaped, (b) isentropic and (c) overhanging. As shown, refrigerants that have the bell-shaped saturated vapor curve do not theoretically require superheating at all, whereas the isentropic require a moderate superheating along with the overhanging require a high superheating. The minimum required superheating (later, ''minSH'') is shown in Figure 6.4 (c) as the temperature increase ''SH'' before the compression to ensure that the refrigerant vapor is superheated after the compression as well. [25,93] In general, the overhanging behavior is linked to refrigerants that have a high molar mass [25]. Note that the shown compression is isentropic. As the compressor isentropic efficiency is lower than unity in the practice (Figure 3.6), there will be a lesser demand for the minSH.

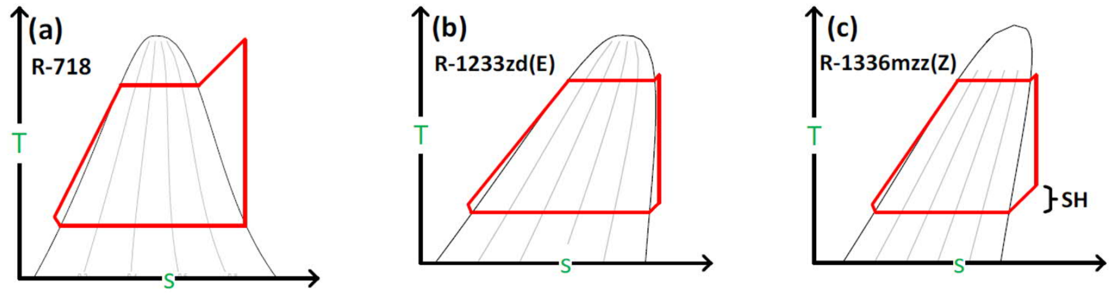

Figure 6.4. Classification of refrigerants on basis the saturated vapor curve [93].

Especially as for the overhanging refrigerants, special superheating and its control is needed to avoid the compression from ending to the two-phase region, i.e. to prevent existence of the refrigerant liquid during the compression (wet compression) and thus the compressor damage. In general, the higher T_{lift} the higher is the minSH. As a solution to prevent the wet compression, use of a so called internal heat exchanger (IHX) is suggested because it is relatively easy to implement and energy efficient. [12,17,24,25,37,93] In practice, IHX can be e.g. a brazed-plate HX [25]. Evidence from IHX practicality gives that several commercialized HTHPs have IHX added to the cycle [17]. The log(p)-h-diagram of an overhanging refrigerant and operating principle of IHX is illustrated in Figure 6.5 that compares the basic cycle to a cycle that is equipped with IHX. The compression ends in the two-phase region if the basic cycle is used (A). To prevent the wet compression, IHX transfers heat from the refrigerant liquid after the condenser by subcooling the liquid and simultaneously, the heat is transferred to the refrigerant vapor after the evaporator to superheat the vapor (B). [25,101] The rightmost side of Figure 6.5 shows the cycle where IHX is added to the basic cycle [17].

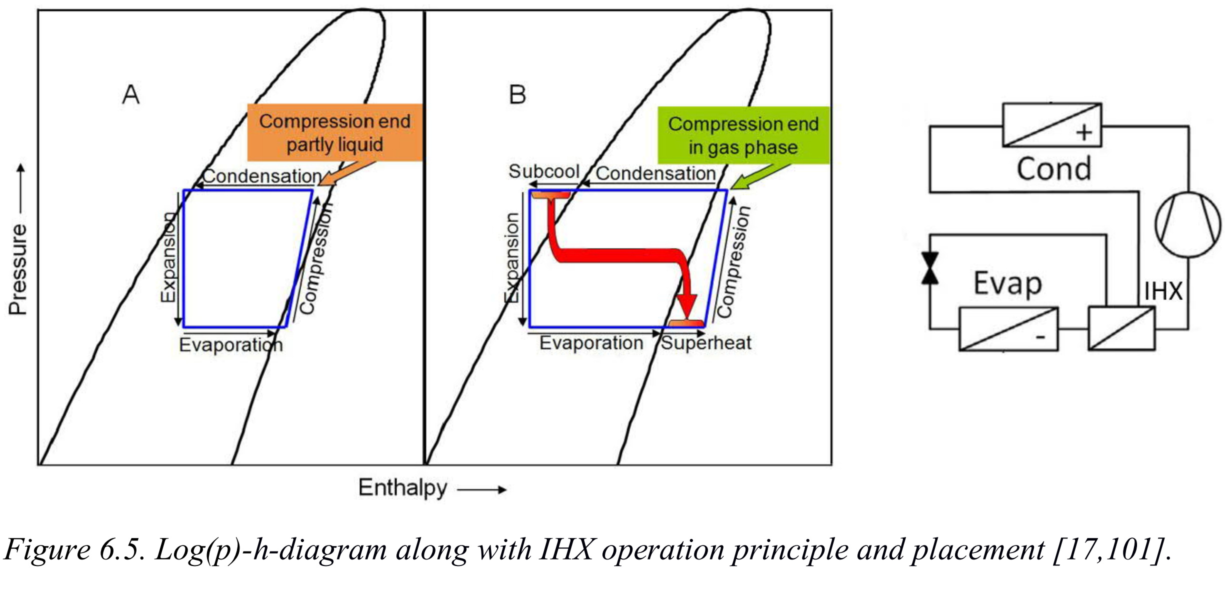

Figure 6.5. Log(p)-h-diagram along with IHX operation principle and placement [17,101].

Conventionally, the superheating is controlled according to the superheat at the exit of the evaporator such that the superheating is few degrees Celsius. Depending on the refrigerant, a minSH can be e.g. 10 °C. This SH implemented with the evaporator could mean a lowering T_{evap} due to a larger HX area serving the superheating and thus increases T_{lift} and decreases both the COP and the VHC [22]. The use of IHX is beneficial due to all the required superheating can be provided with IHX, whereas the evaporator is used to absorb heat from the heat source to evaporate the refrigerant leading to that T_{evap} can be in principle as high as possible with a certain evaporator ΔT_{pinch}. As for cycles with IHX, the superheating at the outlet of the compressor should be used to control the superheating such that it is at least 5 °C. At the same time, it must be ensured that the compressor suction refrigerant is superheated, just like with cycles without IHX. Note that the minimum superheating of 5 °C at the outlet of the compressor is purely from a point of view avoiding the wet compression. [25]

6.2 Compressors

Especially for the higher temperatures, the compressor design and type are important. Every compressor type is not applicable. Often, standard compressors must be adapted for the higher temperatures. Several compressor specifications must be considered when selecting the compressor for HTHPs [12,50] As for closed-cycle vapor compression HTHPs, compressor operating temperatures and pressures are the main limitations. These will define what refrigerant could be suitable. [89] In the light of the mentioned considerations, this section discusses what special technical solutions are required for the compressor and the compression along with discusses the influence of the temperature levels and the pressure levels on the HTHP performance.

6.2.1 Pressure levels

The fundamental considerations when selecting a compressor are the required refrigerant volume flow rate flow rate at the compressor suction and the pressure levels. After the refrigerant volume flow rate has been solved, the needed compressor displacement can be obtained on basis the volumetric efficiency as shown in Equation 9. As for the pressure levels, there is a huge variation across refrigerants that are suitable for HTHPs, thus influencing the compressor choice.

As for HC, HCFO and HFO refrigerants, standard compressors are available for discharge pressures from around 10 to 30 bar and even up to 50 bar. On the other hand, special compressors for natural refrigerants R717, R718 and R744 are available. As for high pressure refrigerants R717 and R744, high pressure compressors suitable for HTHPs are available that can generate discharge pressures even up to 76 bar and at least 140 bar respectively. As for R718, the required compressor displacements are exceptionally high and the pressures are low resulting that large compressors or oil-free high-speed turbo compressors are typically used. [17,29,89,91,92] In addition to the upper limit as for the compressor discharge pressure, the suction pressure is limited and is lower than that of the discharge pressure [24]. The compression pressure ratio should be as low as possible to minimize the compressor electricity consumption, where a practical upper limit could be between

4…8 [17,28,88,89]. However, the pressure ratio of a one compression stage shall not be too small due to the isentropic efficiency could experience a significant drop (Figure 3.6).

The material loads are determined by the pressure levels. In general, the higher pressure level should be kept below 25 bar. [17] However, commercial R717 and R744 HTHPs are available for heat sinks up to around 90 °C and 120 °C respectively (Table 4.1) indicating that there are solutions to withstand the higher pressures.

When combining the refrigerant vapor pressures with the maximum compressor suction and discharge pressures, the maximum achievable T_{evap} and T_{cond} for the combination can be found as exemplary shown in Figure 6.6 for a compressor whose maximum suction and discharge pressures are 10 and 30 bar respectively. As an example, the maximum T_{evap} and T_{cond} with the refrigerant R1234ze(E) could be around 50 °C and 100 °C respectively. [44]

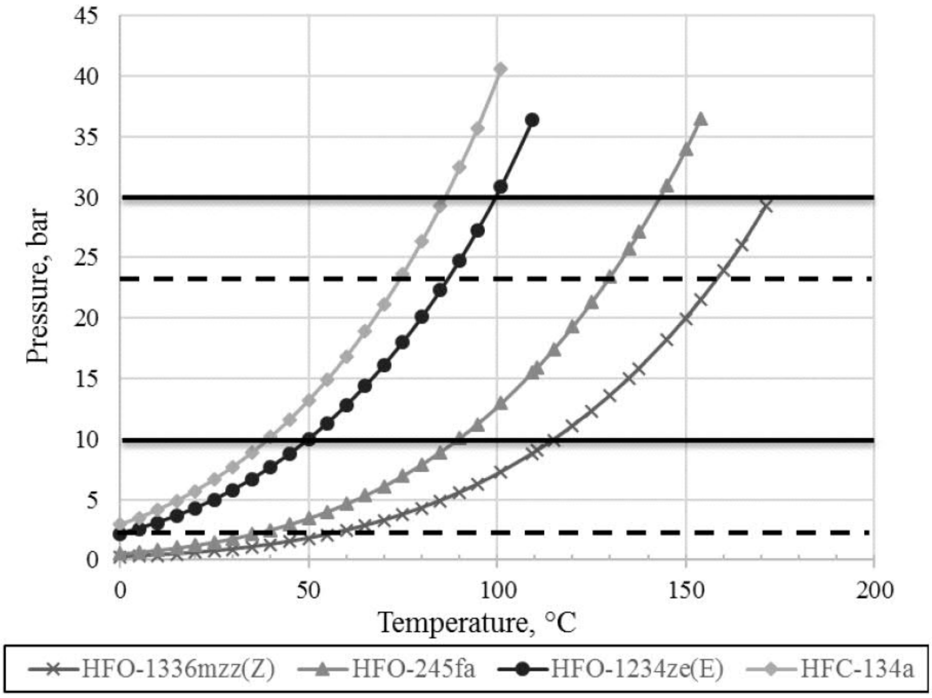

Figure 6.6. Refrigerant vapor pressure curves along with maximum pressures for a compressor [44].

6.2.2 Discharge temperature

The current research provides evidence on that current synthetic refrigerants and compressor lubrication oils cannot withstand too high compressor discharge temperatures. Potential consequences and reasons concerning the high discharge temperatures along with solutions to limit it will be discussed in this section.

To protect the compressor lubrication oil, the compressor discharge temperature must be within a tolerable range. The oil must withstand higher temperatures since typical refrigeration oils can be such that those can start to decompose within the HTHP temperature range. The decomposition reduces the lubrication performance and could damage the compressor. If necessary, an additional compressor cooling or multistage cycles can be applied to keep the compressor discharge temperature within a tolerable level. [12,26,89,91] A condition and a temperature of the compressor lubrication oil influences a lifetime of the bearings, i.e. can influence the maintenance interval and the compressor lifetime [102]. The discharge temperature influences also the wear rate of the compressor mechanical parts, especially at the discharge [28,91]. According to the current research, the compressor discharge temperature must be limited to below 140…190 °C to avoid the decomposition of lubrication oil and synthetic refrigerants along with to reduce the wear rate of the compressor [12,17,24,26,28,29,37,44,88,89,91,92,98,100,103].

The temperature difference between the saturated vapor and compressor discharge temperatures increases when the pressure ratio of the compression increases. Also, the higher ratio of specific heat capacity at a constant pressure to the heat capacity at a constant volume (heat capacity ratio, aka isentropic coefficient), the higher is the increase in the discharge temperature. [12,102] In addition, refrigerants with the bell-shaped saturated vapor curve experience a high superheating during the compression, while the isentropic and the overhanging refrigerants experience a relatively low or none [12]. When compared to the conventional HPs, an excessive high pressure ratio combined with the HTHP condensation temperatures leads easier to excessive high discharge temperatures due to the condensation temperature is closer to the limiting discharge temperatures [104]. Even, when the pressure ratio increases, both the volumetric and the isentropic efficiencies decrease (Figure 3.6). The decrease in the isentropic efficiency raises the discharge temperature even higher due to the enthalpy after the compression gets higher (Equation 10). Thus, the pressure ratio should be limited also in the light of the maximum allowed compressor discharge temperature.

The heat capacity ratio depends on the refrigerant and is relatively high e.g. as for R717 and R718 [102]. For a given fixed operation condition with a fixed T_{lift}, the pressure ratio is a refrigerant dependent being exceptionally high for R718 [17,88,89]. Both R717 and R718 have the bell-shaped saturated vapor curve as well [12]. In the literature, exceptionally high discharge temperatures have been especially noticed for R717 and R718 from which R718 experiences very high discharge temperatures [17,22,24,88,89]. When investigating HCs and the fourth generation synthetic low GWP HCFO and HFO refrigerants, the current research shows that R1234ze(Z) experiences the highest discharge temperatures that are however significantly lower than that of R717 [17,24,88,89,98]. To limit the compressor discharge temperature, multistage cycles are commonly suggested [17,92,98].

As for R718 that exceptionally heats up during the compression, several compression stages with intercooling are needed to keep the discharge temperature at a tolerable level [17,28,29,93]. However, the oil-free turbo compressors could allow higher compressor discharge temperatures and thus can potentially reduce the number of needed compression stages [29,92].

As for R717 when using the basic cycle and supplying heat sinks up to around 100 °C within the limits of R717 compressors, the compressor discharge temperature limit of around 180 °C restricts T_{lift} such that it could be maximally around 60 °C when using the basic cycle. [88,92] On the other hand, application of a 2-stage cycle with an intercooling allows to extend T_{lift} up to around 95 °C where the discharge temperature could be less than 180 °C for R717 [92].

With operating conditions where the heat sink temperature is 110 °C and T_{lift} is around 55 °C, the compressor discharge temperature with HCFO and HFO refrigerants can be between 150…190 °C (190 °C with R1234ze(Z)) when applying the basic cycle equipped with IHX. The discharge temperature can be limited even to 115…125 °C (125 °C with R1234ze(Z)) by applying a multistage cycle. [98] The reduced discharge temperatures were mentioned for the economizer cycle (Figure

3.3) with an exception that there was IHX. Thus, the discharge temperature can be reduced significantly by injecting the refrigerant at the intermediate pressure and by decreasing the pressure ratio. In this case, the discharge temperature could be too high as for R1234ze(Z) without applying a multistage cycle. The discharge temperature can be reduced with other 2-stage or multistage cycles, too [92,98].

The application of IHX can be limited due to its use raises the compressor discharge temperature. The discharge temperature gets higher if IHX is applied because as the enthalpy of the compressor suction refrigerant increases, the enthalpy of the compressor discharge refrigerant increases as well. Thus, the IHX heat transfer effectiveness (IHX heat transfer area) shall not be too high due to the higher effectiveness, the higher is the refrigerant superheating in IHX and the higher could be the compressor discharge temperature. [12,26,28,37,98,100,105]

On the other hand, increasing the discharge temperature has a positive influence on both COP and VHC due to the increasing compressor discharge refrigerant superheating increases the useful heat transfer to the heat sink as the desuperheating increases. However, the discharge temperature must be limited to avoid the overheating issues. [26,37,98,100,105] As for multistage cycles, the intermediate stage refrigerant intercooling can be implemented with a desuperheater that cools down the refrigerant and heats the heat sink, especially in case of refrigerants that experience high discharge temperatures, such as R717. The desuperheating with a condenser, a separate desuperheater or an intercooler before the condensation allows to reduce the condensation temperature, thus reduces T_{lift} and so raises COP. [21,59,106]

To limit the compressor discharge temperature and the lubrication oil temperature, an additional cooling with e.g. water can be applied. Usage of a separate oil cooler is suggested for HTHPs to ensure enough low oil temperatures to increase the operating life of compressors. [28,106] Avoiding an additional wear rate of the compressors should be indeed considered since the compressors are the major investment cost of HTHPs.

In addition to the compressor discharge temperature, the compressor suction temperature shall not be excessive high, especially as for hermetic or semi-hermetic compressors where the suction refrigerant cools the electrical motor of a compressor. As for the suction refrigerant cooled compressors, the maximum suction temperature could be 80 °C to allow an effective cooling of the motor, the compressor casing and the lubrication oil. The heat generation at the motor can be reduced by sizing the motor e.g. 25 % larger than would be necessary. This limits the suction refrigerant temperature rise due to cooling of the motor. As for the suction refrigerant cooled compressors, the IHX heat transfer effectiveness is further restricted. [28,50,89,97,100]

7 Practical technologies

This section discusses the most promising and practical HTHP technologies that were found during the survey. The technologies are divided into the refrigerants, the compressors and the cycle configurations along with they are discussed in detail in touch with their own subsections.

7.1 Refrigerants

This section provides a detailed discussion on both environmentally friendly and probably future-proof refrigerants that are suitable for HTHPs within the limitations of this thesis. In the light of the legislation and the current research, it is evident that only refrigerants that are not ODS and have as low GWP as possible should be used in the future. In the end of this section, the most promising current refrigerants are concluded for the target heat sink temperature range. Note that refrigerant mixtures were not considered in this thesis. Those should be potentially tailored for different projects and thus probably decreases the practicality. As shown in section 4, mixtures are not generally used in current HTHPs. However, the COP improvement potential is notable when compared to pure refrigerants as shown in section 3 and e.g. in [16,103].

7.1.1 Properties and suitability

This section discusses practical and the most suitable environmentally friendly refrigerants for the target heat sink temperature range described in section 1.1. From the discussed refrigerants, several are commonly used in the current commercialized HTHPs (Table 4.1) or in other applications that are related to the refrigeration technology. Across the current research, the discussed refrigerants are the state-of-the-art [17,21,24,26,88,89,98,100,107].

Table 7.1 summarizes the most important cycle-independent properties of the potential HTHP refrigerants extended with saturated vapor pressures at 40 °C (heat source) and at 90 °C (heat sink) along with the pressure ratio resulting from those. The refrigerants are sorted according to their type along with T_{crit} from a lower to a higher. In general, when T_{crit} gets higher, the NBP gets higher. When T_{crit} is around 170 °C, the NBP gets over 20 °C meaning that with this ambient air temperature, a higher T_{crit} of around 170 °C can cause a risk of an air leakage to the HTHP system during down-times. A low T_{crit} seem generally to lead high operating pressures. In general, p_{crit} are moderate being from around 30 to 50 bar, but the natural refrigerants have exceptionally high p_{crit}. With the exemplary conditions, the pressure ratio seems to increase with increasing T_{crit}. Most of the refrigerants are at least mildly flammable. In general, the larger molar mass, the higher superheating could be needed according to the shape of the saturated vapor curve. However, HCs tend to have the overhanging behavior already with smaller molar masses. These trends linked to the molar mass and shape of the vapor curve are confirmed in [25]. According to the shape of the saturated vapor curves, most of the refrigerants could require a special superheating (e.g. IHX) since the superheating demand could be moderate or high.

Table 7.1. Potential HTHP refrigerants.

	refrigerant	T_{crit} [°C]	p_{crit} [bar]	ODP [-]	GWP [-]	SC [-]	NBP [°C]	M [g/mol]	$p_{sat,40}$ [bar]	$p_{sat,90}$ [bar]	p_{ratio} [-]	shape [-]	reference
HC	R1270	**91**	46	0	2	**A3**	-48	42	16.5	**44.7**	2.7	bell	[17,108]
	R290	**97**	43	0	3	**A3**	-42	44	13.7	**37.7**	2.8	bell	[17,108]
	R600a	135	36	0	3	**A3**	-12	58	5.4	16.7	3.1	is	[17,108]
	R600	152	38	0	4	**A3**	-1	58	3.8	12.4	3.3	is	[17,108]
	R601a	188	34	0	4	**A3**	**28**	72	1.5	5.8	3.9	over	[17,109]
	R601	197	34	0	4	**A3**	**36**	72	1.2	4.7	3.9	over	[17,109]
natural	R744	**31**	74	0	1	A1	-78	44	**supercritical**			bell	[89,108]
	R717	132	113	0	0	**B2L**	-33	17	15.5	**51.1**	3.3	bell	[17,108]
	R718	374	221	0	0	A1	**100**	18	**0.07**	**0.7**	**10.0**	bell	[17,108]
HFO	R1234yf	**95**	34	0	4	**A2L**	-29	114	10.2	**30.8**	3.0	is	[98,110]
	R1234ze(E)	109	36	0	1	**A2L**	-19	114	7.7	24.8	3.2	is	[98,111]
	R1336mzz(E)	138	32	0	18	A1	8	164	3.2	12.2	3.8	over	[90,98]
	R1234ze(Z)	154	36	0	1	**A2L**	10	114	2.9	10.8	3.7	is	[98,111]
	R1336mzz(Z)	171	29	0	2	A1	**33**	164	1.3	5.6	4.3	over	[98,112]
HCFO	R1224yd(Z)	156	33	0.0002	1	A1	15	149	2.5	9.3	3.7	is	[98,113]
	R1233zd(E)	166	36	0.0003	1	A1	18	131	2.2	8.3	3.8	is	[98,114]

Unfavorable properties are highlighted with a bold red color in Table 7.1. A lower T_{crit} than around 110 °C are highlighted due to a too low T_{crit} causes that the subcritical operation is not efficiently possible for heat sinks of 80…100 °C. However, e.g. T_{crit} of around 95 °C could be suitable for heat sinks up to around 80 °C because there could be a reasonable condensation enthalpy remaining. The safety classes that differ from A1 (non-toxic and non-flammable) are highlighted because those could require necessary safety measures. NBP of 20 °C and higher are highlighted due to at least the HTHP low-pressure side could be at vacuum conditions with heat sources and ambient air that are at temperatures below 20 °C and thus, there could be the risk of the ambient air leakage. Higher vapor pressures than 30 bar were highlighted due to increasing material loads along with majority of compressors (except R717 and R744) are not able to work within a such high discharge pressures.

As for HCs, the main challenge is their high flammability. However, this is acceptable in some conditions according to SFS-EN 378-1:2016 [36] when necessary safety measures are considered. R1270 and R290 have a relatively low T_{crit} indicating that they are not suitable for heat sinks up to 100 °C. Those could however be suitable in lower temperatures. On the other hand, those have high vapor pressures indicating higher VHC but also higher material loads. R601 and R601a have a relatively high T_{crit} leading to higher pressure ratios along with a high NBP and low vapor pressures indicating low VHC and potential air leaks in some conditions. R600 and R600a have a moderate T_{crit} and a low NBP potentially indicating that both COP and VHC could be good. Due to R1270 has a lower T_{crit} than 95 °C, it is not considered further in the HTHP temperature range. Despite of the flammability of HCs, those are suggested as a one HTHP refrigerant according to current research along with the compressor technology as for HCs is available [17,21,24,27,28,50,89,115].

R744 could require a supercritical operation in the exemplary operating conditions or at least transcritical operation if the heat source could be at a lower temperature. The pressures could be very high in the conditions, around 100 bar for the higher pressure side [21,116]. R744 is not suitable for the subcritical operation due to a too low T_{crit}. According to the current research and the currently commercialized HTHPs, R744 has been found to be especially efficient in the transcritical operation where the heat source temperature is at least moderately below T_{crit} and the heat sink experiences a very high temperature glide, e.g. from 20 to 100 °C [17,21,28,50,64,89,116,117]. As for transcritical HTHPs, the efficiency improvement is based on that the cycle partly behaves according to Equation 4. This is a more efficient when compared to Equation 3 that is valid for subcritical HTHPs operating with pure refrigerants. Despite of the potential efficiency improvement with the high temperature glides, R744 will not be considered further because the subcritical operation is not possible in the target temperature levels along with the suitability for the high temperature glides potentially restrict the applications where R744 could be efficiently used.

As for R717, the pressure is not a too bad challenge currently since compressors are available. Due to R717 experiences relatively high compressor discharge temperatures and thus desuperheating, its T_{lift} can generally be lower when compared to other refrigerants to achieve a required $T_{sink,out}$. However, required safety measures must be considered due to R717 is both toxic and mildly flammable. In general, those safety measures have not been considered to be too heavy due to R717 is a relatively common and well known HTHP refrigerant (Table 4.1) and has been used in various installation locations such as in the industry and in urban areas [17,48,50,56,106,118]. In addition, it has been found that R717 is more efficient than R744 at least with moderately high heat sink temperature glides [119,120]. As the vapor pressures of R717 are high, it is expected that the VHC is high along with the pressure ratio seems to be quite low indicating good compressor efficiencies.

As for R718, both the vapor pressures are very low and below 1 atm indicating that the whole system could be in vacuum conditions along with the pressure ratio is very high. However, R718 is an efficient refrigerant especially for heat sources of at least around 100 °C and higher because its VHC gets higher due to higher vapor pressures along with COP gets higher with increasing $T_{sink,out}$ due to T_{crit} does not restrict the performance within the higher temperatures [40,92,88,121].

HFOs seem to be promising since the vapor pressures are moderate or relatively low along with those are only mildly flammable or not at all. However, R1234yf could only be suitable for heat sinks up to around 80 °C due to a relatively low T_{crit}. In addition, R1336mzz(Z) has NBP of 33 °C meaning the risk of the ambient air leaks in some conditions.

HCFOs are promising like HFOs along with HCFOs are not flammable at all. However, HCFOs have a low ODP that was analyzed. According to the current literature, it seems to be such that the use of HCFOs is not restricted since HCFOs decompose in the lower troposphere and thus unlikely deplete the stratospheric ozone layer, HCFOs are not controlled by the Montreal Protocol (Kigali Amendment) along with HCFOs are actively researched and are often considered as good options, manufactured, marketed and sold [25,33,37,50,93,97,98,101,105,113,114,122,123].

7.1.2 Performance and the most promising options

In the following, the most important performance indicators COP and VHC are compared across the potential refrigerants to distinguish the most promising options. The comparison is done for heat sinks at least up to 100 °C. As for the comparison, simulation results from [88] were used as a base, where the basic cycle was applied with $T_{sink,out}$ of 90 °C and $T_{source,out}$ of 31 °C along with ΔT_{pinch} was 5 °C for HXs resulting T_{lift} of around 60…70 °C depending on the refrigerant. The simulation had some assumptions such as η_{is} of 0.80 and pressure losses in the HXs were neglected. However, the relative performance was considered to be important. Similar simulations were available from [17,24,25,89,96,115] having moderate correlation giving evidence for the comparison.

Figure 7.1 compares COP and VHC across the refrigerants. From the previous section, R744 and R1270 were excluded from the comparison. As for most of the refrigerants, the simulation results were available from [88] and some were extrapolated by using data from few sources as follows. Both R1224yd(Z) and R1336mzz(Z) were extrapolated from [97] as those were not available in [88] but both those were available in [97] along with R1233zd(E) was utilized for the extrapolation since it was available in both the references [88,97]. Similarly, R1234yf and R1336mzz(E) were extrapolated from [90] as those were not available in [88] but both those were available in [90] along with R1234ze(E) was utilized for the extrapolation since it was available in both the references [88,90]. The operating conditions were similar in all of those references. In addition, the current conventional and commonly used high GWP HFCs R134a and R245fa were available from the base [88] and were used as a reference. Note that those cannot be recommended as sustainable options since those are currently controlled according to the F-gas regulation [34] and the Kigali Amendment [39].

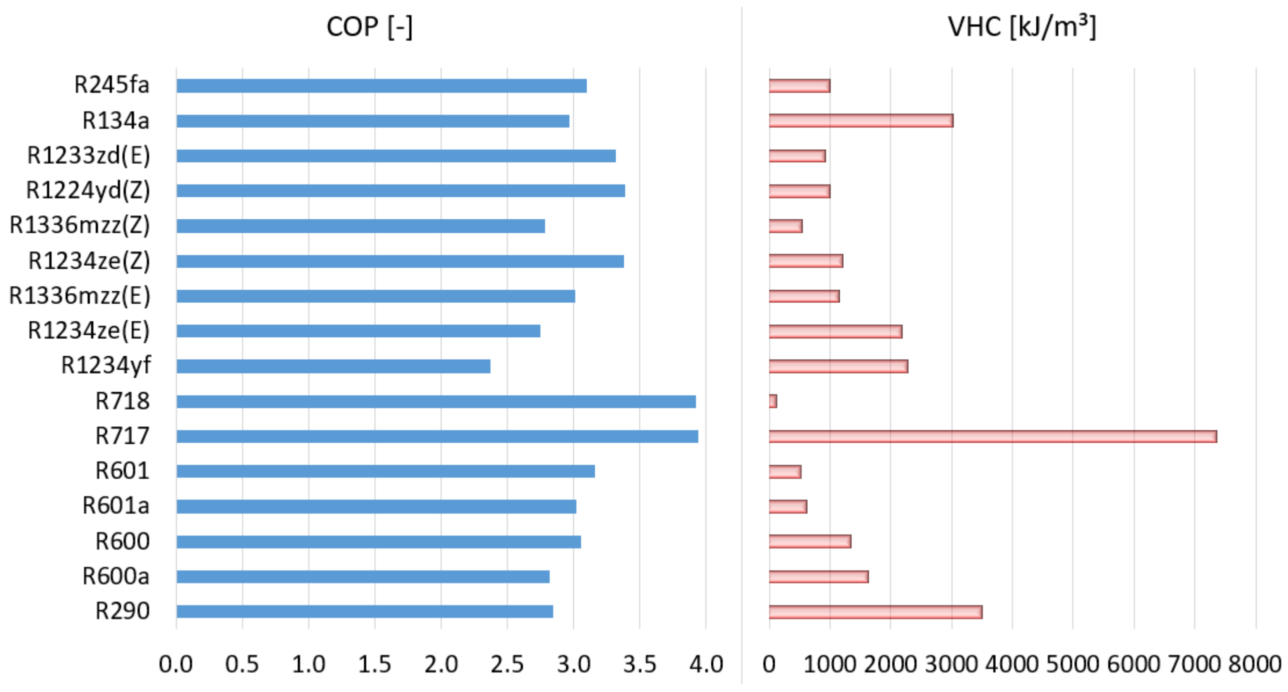

Figure 7.1. Comparison of COP and VHC across potential refrigerants.

As the first observation from Figure 7.1, R717 should be used whenever possible at least for heat sinks up to 90 °C because it has both the highest COP and the highest VHC along with the compressor technology is available. However, the safety measures must be considered and those can restrict the use in some conditions. R718 has roughly the same COP with R717 but, the VHC of R718 is clearly the lowest. Due to the very low VHC and vapor pressures of R718 with the target heat sink temperatures, R718 is not recommended since additional solutions could be needed for the low pressures along with the compressors could be very large and expensive. In addition, the very low VHC indicates potential reduction in practical COP according to Figure 6.3 since the relative compressor electricity consumption increases when the VHC decreases.

To proceed, R1234yf has the lowest COP and a moderately high VHC, thus it is not recommended for heat sinks of 90 °C or higher since the COP reduction due to too low condensation enthalpy could be a too high. For heat sinks of around 80 °C, its COP can be still relatively low and is not recommended. As for R1336mzz(Z), its COP is relatively low and its VHC is clearly lower than 1000 kJ/m^3. R1336mzz(Z) was extrapolated from an experimental study to highlight that a too low VHC decreases the practical COP. If the COP of R1336mzz(Z) was extrapolated from a theoretical simulation, e.g. [90], it could be higher than that of R245fa in that case. However, it was concluded in [97] that R1336mzz(Z) has a too low VHC for heat sinks of 80…100 °C indicating reduction in the practical COP and relatively large compressors could be needed and is not thus recommended. For higher heat sink temperatures, R1336mzz(Z) could achieve good COPs and moderate VHCs. When considering the enough VHC for the target heat sink temperatures, R601 and R601a should not be used since it is probable that their VHC are also too low and thus the showed COPs could be lower in practice. Also, NBP of R601 and R601a is so high that it restricts to use lower temperature heat sources since the low pressure side of the system could be in the vacuum conditions. Moreover, R290 shows a moderate COP and the secondly highest VHC. However, the vapor pressures are relatively high, especially at the high pressure side that can potentially restrict the use of R290 for heat sinks of 90 °C. With T_{cond} of 80 °C, the vapor pressure could be still around 31 bar [108] and could be at a limiting pressure for HC compressors. Thus, R290 could potentially be used for lower temperature heat sinks of around 80 °C or a little higher where its COP and VHC could be competitive.

As for the remaining refrigerants, all of those could be suitable for heat sinks up to 100 °C according to their T_{crit} from which R1234ze(E) has the lowest COP and lowest T_{crit} and thus experiences already a decreasing COP and decreases more with higher temperature heat sinks. However, it could be a very effective for heat sinks of at least 90 °C since there is a sufficient temperature difference to T_{crit} and its VHC is the highest from the remaining options. In addition, R1234ze(E) is an extensively used HFO in the current commercialized HTHPs for heat sinks even up to 100 °C (Table 4.1) giving practical evidence from its applicability. While the other options have VHC of around 1000…1500 kJ/m^3 for heat sinks of 90 °C, it can be that those could experience some reduction in the practical COP if those could be applied for heat sinks lower than 90 °C. On the other hand, both R600 and R600a have a moderate VHC for heat sinks of 90 °C and a higher COP than R1234ze(E), thus those could be also good options for lower temperature heat sinks when a higher COP is important.

Finally, the most suitable remaining refrigerant at the heat sink temperature of 100 °C, could be as follows. According to experiments from [97], R1224yd(Z) could be slightly better refrigerant in terms of both COP and VHC for heat sinks of at least up to 100 °C when compared to R1233zd(E) due to the practical influence of VHC on COP as shown in Figure 7.1. Moreover, R1234ze(Z) could be even somewhat better than R1224yd(Z) because it has the same COP and slightly higher VHC. Both R1336mzz(E) and R600 seem to achieve worse performance than both, R1234ze(Z) and R1224yd(Z). The best remaining options R1234ze(Z) and R1224yd(Z) for heat sinks of 100 °C are similar except R1234ze(Z) is mildly flammable whereas R1224yd(Z) is not flammable at all. Thus, if the mild flammability is not an issue, R1234ze(Z) should be the choice along with if the flammability is an issue, R1224yd(Z) should be used.

As a summary, the most promising refrigerants could be like prioritized below from top to bottom.

- Use R717 whenever possible if compressors are available with respect to pressures;
- If R717 cannot be used, use R1234ze(Z) for heat sinks at least from 90 to 100 °C;
- If R1234ze(Z) cannot be used, use R1224yd(Z) for heat sinks at least from 90 to 100 °C;

- For heat sinks from 80 to 90 °C, use R1234ze(E);
- For heat sinks from 80 to 90 °C, use R600 or R600a if a higher COP when compared to R1234ze(E) is important and the high flammability is not an issue;
- For heat sinks of 80 °C, use R290 if compressors are available and the high flammability is not an issue.

Note that the comparison was solely based on the COP and the VHC shown in Figure 7.1 along with on the potential negative influence of the too low VHC on the practical COP. Note that the relative trends of the COPs and the VHCs could change if the heat source and the heat sink temperatures change. If $T_{sink,out}$ is kept constant and T_{lift} is varied, both the COP and the VHC vary from which especially the variation in the VHC causes uncertainty for the comparison since when T_{evap} reduces, the VHC reduces due to e.g. the reducing refrigerant density at the compressor suction [24,90]. Thus, with lower T_{lift} than the 60…70 °C that was valid in the comparison, the VHC could be higher with all of the refrigerants indicating that the VHC of e.g. R1234ze(Z) and R1224yd(Z) could be reasonably high for the whole considered heat sink temperature range from 80 to 100 °C. To find out more accurately the most promising refrigerants within certain conditions, a lifecycle analysis on the costs and potential emissions ($TEWI$) should be done like presented e.g. in [37,91,92,103,104]. According to this a more holistic approach, R717 can perhaps be the cost-optimal choice like concluded in [91,92]. However, a potential premium cost of R717 systems shall not be too high when considering the economical profitability. In addition, R717 could minimize $TEWI$ due to the highest COP and zero GWP. But, a more accurate combined cost and emission analysis is strongly suggested to find out the optimal options within certain target heat source and heat sink temperatures.

7.2 Compressors

This section discusses current commercialized compressors that are or could be suitable for HTHPs within the considered limitations and suggestions this far. Only 1-stage compressors were considered. However, it was checked that 2-stage compressors are available at least from 6 manufacturers that are mentioned in this section. Uncertainty for the compressor suitability was noticed to originate from that several manufacturers do not announce the new synthetic low GWP HFO and HCFO refrigerants in their technical data. Despite of this uncertainty, the pressure levels and the maximum allowable discharge temperature were considered to determine suitable compressors like considered e.g. in [17,89,91]. Compressors that cannot withstand the compressor discharge temperatures of at least 120 °C were not considered. As for heat sinks up to 100 °C, this can be considered as a precondition [89].

According to the survey, commercialized piston and screw compressors were found that could be suitable. This is in an agreement with capacity range presented e.g. in [22,29]. However, only the piston compressors were considered as for R717 since available screw compressors seem to be suitable for too large heating capacities. The listed pressures are the maximum compressor operation pressures. Compressor design pressures were typically around 10 % higher than the maximum operation pressures. Both the maximum suction pressure $p_{in,max}$ and the discharge pressure $p_{out,max}$ were listed. The listed volume flow rates $\dot{V}$ are the compressor displacements. As for R717 compressors, $\dot{V}_{min}$ is given at a minimum announced adjustable speed with VSD and $\dot{V}_{max}$ is given at a maximum announced speed. As for the other compressors, both displacement limits are given at a maximum speed since there were lots of models with different displacements available.

Commercial compressors that were found are summarized in Table 7.2. An informative illustration from the compressor displacements and the pressures is shown in Appendix 4. All in all, 18 models were found from 9 manufacturers. The compressors are basically classified according to the compressor technology and the refrigerant suitability to three classes, where R717 has its own class. On the other hand, the other suitable refrigerants are classified for either the piston or the screw technology. Majority of the suitable compressors seem to be pistons that are available for displacements up to around 1350 m^3/h, while suitable screws seem to be available from around 80 to 11200 m^3/h. The maximum operation pressures seem to deviate considerably across the compressor models in each class. As for R717 compressors, the maximum suction and discharge pressures vary from 16 to 30 bar and 45 to 60 bar respectively. As for other refrigerants, the pressures vary as for pistons from 6 to 21 bar and 16 to 32 bar respectively along with as for screws from 19 to 28 bar and 25 to 52 bar respectively. Note that the maximum suction pressure was not available for one screw compressor. In addition, the maximum compressor discharge temperature along with the minimum and the maximum pressure ratios were announced with some compressors. The maximum discharge temperature varied from 120 to 160 °C. The minimum pressure ratio was found to be least 1.5. The pressure ratio shall not exceed 5…10 for the piston compressors along with the upper limit for the pressure ratio was given to be even 22 for GEA Refrigeration screw compressors.

Table 7.2. Commercialized compressors.

manufacturer	product	HFO=HFO&HCFO compressor	refrigerant	V_{min} [m^3/h]	V_{max} [m^3/h]	$p_{in,max}$ [bar]	$p_{out,max}$ [bar]	reference
GEA Refrigeration	Grasso M, LT	screw	HC,HFO	231	11244	28	52	[58]
J&E Hall	HSO	screw	HC,HFO	175	2486		40	[124]
Johnson Controls	CMO, SMC	piston	HC,HFO	97	1357	16	25	[59]
Bitzer	CS	screw	HC,HFO	137	1015	19	28	[125,126]
Johnson Controls	Frick RXF	screw	HC,HFO	122	1013	21	25	[127]
Mayekawa	WBHE	piston	HC,HFO	250	770	6	24	[52]
Mayekawa	HS	piston	R717	200	600	30	60	[52]
Carlyle	5F, 5H	piston	HFO	34	588	6	16	[128]
Johnson Controls	HPX	piston	R717	37	443	24	55	[59]
Bitzer	HS	screw	HFO	84	410	19	28	[125,126]
Mayekawa	WA	piston	HC,HFO	80	360	6	20	[52]
Bitzer	ECOLINE	piston	HC,HFO	4	303	19	32	[125,126]
GEA Refrigeration	HGHC, HG	piston	HC,HFO	5	281	19	28	[58,129]
Dorin	H, HEX	piston	HC,HFO	4	245	6	26	[130]
Frascold	AXH, AY…WY	piston	HC,HFO	4	239	21	30	[131]
GEA Refrigeration	Grasso HP	piston	R717	29	202	16	45	[58,132]
Mayekawa	HK	piston	R717	120	200	20	50	[52]
Viking Heat Engines	HBC 511	piston	HFO	14	42	10	30	[43]

Figure 7.2 shows the working domains of the compressor technologies listed in Table 7.2 bounded with lines. The working domains for different refrigerants are shown with hatched areas where pressures are shown for heat sinks from 80 to 100 °C and the compressor displacement demands for

heating capacities from 50 to 1000 kW so that one compressor is assumed in HTHP. R717 pressures for heat sinks of 80 and 100 °C were checked from [108] and [89] respectively. As for other refrigerants than R717, an extreme for low pressure refrigerants is shown with R1233zd(E) whose pressures were checked from [114] and [89] respectively, whereas the extreme for high pressure refrigerants is shown with R1234ze(E) whose pressures were checked from [111] and [89] respectively. The installed compressor displacements were gathered from [88] at 90 °C heat sink and at 1000 kW heating capacity where 80 % compressor volumetric and isentropic efficiencies were assumed. Smaller displacements were extrapolated for 50 kW heating capacity by using Equation 1 and by assuming that the cycle conditions stay the same in other matters. As for R717, the displacements from [88] were verified by using manufacturer datasheets from [52,59]. As for other refrigerants, the displacements were verified for R1234ze(E) by using manufacturer data from [125].

Figure 7.2 gives an idea how the commercialized compressors suit the target heating capacities with the suitable refrigerants. As for R717, majority of the target domain could be achieved with the commercialized compressors. The heat sinks of 100 °C could potentially be achieved with the R717 compressors due to T_{cond} is 98 °C at 60 bar vapor pressure [108] and the desuperheating is significant with R717 depending however on the cycle conditions. R717 compressors seems to have a mismatch at small heating capacities. Heat sinks up to around 90 °C can be achieved with R717 compressors from around 100 to 500 kW, while heat sinks up to around 100 °C can be achieved for 500 kW and larger heat sinks. Note that the smallest heating capacities require a compressor capacity control e.g. with VSD. Comparison with the heat sink temperatures listed by compressor manufacturers and HTHP manufacturers (Table 4.1) shows evidence that heat sink temperature of 95 °C could be at least achieved with the current commercialized standard R717 piston compressors.

As for other refrigerants than R717, the target heating capacities and the target temperatures could be achieved with the compressors. However, there is a small mismatch for the screw compressors with smaller heating capacities and high pressures along with the piston compressors have a mismatch for larger heating capacities, especially due to R1234ze(E) needs a relatively high pressure for 100 °C heat sinks. As for the piston compressors, the mismatches can be solved by using several parallel installed compressors to increase the HTHP heating capacity [17]. The parallel installed pistons could be needed especially as for R1234ze(E) due to the only suitable compressor for 100 °C heat sinks could achieve around 200 kW heating capacity [125].

As shown in Figure 7.2, the higher refrigerant vapor pressure, the smaller compressor displacement demand and thus the smaller refrigerant volume flow rate and a higher VHC. This can be confirmed from Figure 6.2 along with by checking Table 7.1 and Figure 7.1. Thus, the working domains of the other suitable HCs, HFOs and HCFOs are between the extremes R1234ze(E) and R1233zd(E) shown in Figure 7.2. Moreover, R717 seem to require very small displacements when compared to other the refrigerants. Exceptionally high VHC of R717 explains this [17,88,89]. This means smaller compressors along with potentially smaller investment costs for compressors if the higher pressures do not result in a premium cost [17,91]. The compressor working domains and R1233zd(E) displacement demand in Figure 7.2 gives a moderate agreement when comparing with similar domains in [29]. Note that the displacement demand depends on the cycle conditions, especially from T_{lift} such that the displacement demand increases with increasing T_{lift} [29,50]. The displacements in Figure 7.2 are for the basic cycle with a relatively high T_{lift} of around 60…70 °C depending on the refrigerant. Thus, the displacement demand for the target heating capacities could be lower with lower T_{lift} being possible with higher heat source temperatures if the heat sink temperature is fixed.

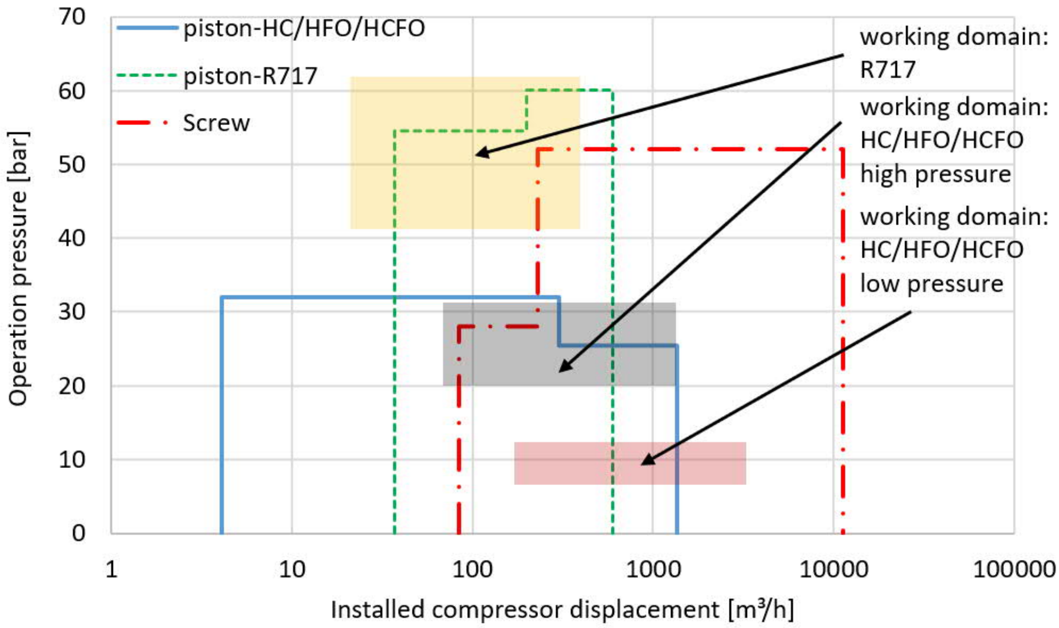

Figure 7.2. Working domains of current commercialized compressors and refrigerants.

According to the survey, it is evident that the piston compressors are optimal for R717 HTHPs within the considered heating capacities. According to basic manufacturer data, it is not clear whether to select the piston compressor or the screw compressor as for the other suitable refrigerants. Efficiency of capacity control can be considered as a one important compressor feature since it can be that a heat demand of an application varies, even significantly. As for the R717 piston compressors, the maximum displacement of even the smallest compressors is too large for smaller heating capacities meaning a demand to adjust the displacement, i.e. the speed, preferably with VSD. On basis GEA RTSelect and VAP along with Bitzer software [125,129,132], COP with piston compressors is roughly constant when the speed is varied from the maximum towards smaller. As for R717 pistons (GEA), the COP could even slightly improve when the speed reduces. On the other hand, COP with screws seem to reduce significantly when the speed is reduced from maximum to lower speeds.

Piston compressors typically have a lower energy consumption when compared to screw compressors with heating capacities lower than around 1200 kW. If compressors must operate with a part load often, the piston compressors are the optimum choice because a better part load efficiency. Piston compressors can adapt naturally to varying evaporation and condensation temperatures, whereas small screw compressors are typically made with fixed pressure ratios. In a long run, piston compressors could have around 20…30 % higher maintenance costs, but the operational energy savings when compared to the screw compressors are typically several times larger. [133]

According to the discussion between piston and screw compressors, the piston could be the optimal choice for the target HTHP operation range due to better adjustability, efficiency and a larger division of available models. Too small piston displacements for larger heating capacities can be solved by using several parallel installed compressors. This arrangement can be beneficial also for smaller heating capacities to obtain reliability for HTHP along a better capacity adjustability could be achieved. Thus, if one compressor has some errors, HTHP could operate with the remaining functioning compressors instead of a total end of the operation. Parallel installed compressors are a common practice when piston compressors are used in the current commercialized HTHPs [17]. An additional evidence for the piston compressors gives that the current commercialized HTHPs with piston compressors are generally up to heating capacities of around 1000 kW (Table 4.1).

7.3 Cycle configurations

This section discusses the most promising practical cycle configurations and for a which T_{lift} range those could be suitable. The focus was only in the main components as mentioned in section 1.1. The cycles were sorted into two groups according to whether the refrigerant is the high pressure R717 or the other moderate pressure refrigerants HC, HCFO and HFO. For these groups, such cycle configurations are suggested that are recognized to be both effective and practical according to the survey. Cycles that have only up to two compression stages were considered since the heat source inlet temperature to a HTHP could be minimally around 5 °C (Figure 5.2 & Figure 5.3) and the maximum target heat sink temperature is 100 °C meaning that T_{lift} could be around 100 °C depending on ΔT_{pinch} for HXs, the amount of superheating in the evaporator and the amount of desuperheating before condensation. There are some 2-stage cycles that could perform reasonably well for T_{lift} up to even 120 °C where T_{lift} per a compression stage could be around 60 °C [26,50].

The compressor discharge temperature restricts a too high T_{lift} from one compression, especially as for R717 as discussed in section 6.2.2. As for the suggested cycles, the maximum discharge temperature was considered to be 180 °C for all compressors since this temperature is the most common in the current research. As for the potentially suitable compressors (Table 7.2), the maximum discharge temperature was 160 °C for some compressors. The gap in the temperature can be potentially solved with a compressor manufacturer [28]. To ensure that the discharge temperature is limited and potentially kept well below the limiting temperatures, it is recommended to use an extra lubrication oil cooler, especially as for R717 [106].

For all 1-stage cycles, the minimum T_{lift} was determined according to the minimum pressure ratio of 1.5 that was found from some compressor manufacturers. By using the pressure ratio, T_{lift} was checked from [108] for various refrigerants resulting a minimum T_{lift} of around 15 °C. According to [74], the minimum T_{lift} is suggested to be 20 °C. On the other hand, pressure ratio of 1.5 could already decrease the compressor isentropic efficiency (Figure 3.6). Thus, the minimum T_{lift} is suggested to be at least 20 °C to ensure that the pressure ratio is enough high.

There is a large assortment of cycle configurations that vary in a cycle complexity and performance utilizing either 1- or 2-stage compression [17,24,26,30,31,94,96,98,100,104,117,134,135]. In the following, the suggested cycles from the assortment are first discussed and secondly, the cycle configurations and their log(p)-h-diagrams are illustrated. Only the suggested cycles are discussed since there are by the scores of possible cycles that could require an excessively broad discussion.

The suggested practical 1-stage cycle for R717 is shown in Figure 7.3 for T_{lift} from 20 to 60 °C. The cycle is the basic cycle that was gone through already in detail in touch with Figure 3.1. Practical evidence gives that this cycle configuration is applied in several current commercialized R717 HTHPs in touch with a moderate T_{lift} up to around 60 °C [17,59]. IHX has been noticed to significantly improve both COP and VHC as for most of the refrigerants [12,26,88,97,98]. However, there is scientific evidence that IHX should not be used as for R717 because it does not benefit from the usage of IHX along with IHX can even decrease both COP and VHC [12,88]. A higher T_{lift} than 60 °C is not potentially feasible with 1-stage compression since the compressor discharge temperature could be a too high [12,88,92] along with the pressure ratio could increase leading to decreasing compressor efficiencies.

The suggested practical 2-stage cycle for R717 is shown in Figure 7.4 for T_{lift} from 60 to 95 °C. The cycle utilizes a so called flash tank (Flash) to separate the evaporating refrigerant vapor (flash gas) from the liquid during the first expansion (4→5). Thus at the intermediate pressure, the saturated vapor (2b) is separated from the saturated liquid (6) inside the flash tank after which the liquid is expanded again (6→7) and evaporated in the evaporator (7→1). After the evaporation, the first compression is implemented (1→2a) after which the saturated vapor (2b) is injected to intercool the superheated vapor (2a→2) and is compressed to the condensation pressure (2→3) along with the heat is supplied to the heat sink (3→4). Both COP and VHC increases with this cycle when compared to the basic cycle because the expansion losses are partly recovered (enthalpy difference 4-7) along with the compression is implemented in two stages that allows a higher T_{lift} and better compressor efficiencies. In addition, the compressor discharge temperature could be limited below 180 °C with heat sinks up to around 100 °C due to a higher isentropic efficiency and the usage of intercooling with T_{lift} up to around 95 °C without extra lubrication oil cooling [92]. Practical evidence for this cycle gives that one of the current commercialized R717 HTHPs has a similar 2-stage cycle for T_{lift} up to around 90 °C resulting Carnot efficiency even up to around 0.7 [59]. According to the survey, there were no R717 compound 2-stage compressors. Thus, two separate compressors should be applied from which the low stage compressor can be a low pressure R717 compressor with higher displacements due to a smaller vapor density, whereas the high stage compressor must be a high pressure R717 compressor with smaller displacements due to a larger vapor density [59,108]. Scientific evidence for this cycle gives that the performance is slightly better than that of the economizer cycle (Figure 3.3) along with this flash tank cycle is commonly applied and suggested in the research [30,92,98,104,107,134-136]. An improvement when compared to the economizer cycle is that the flash tank cycle can potentially achieve higher evaporation enthalpy leading to higher COP and VHC. Note that in general for every refrigerants and 2-stage cycles, 1-stage cycle should be used for lower T_{lift} since the potential improvement from a 2-stage cycle could potentially be very small with lower T_{lift} and further, with T_{lift} of around 50 °C and lower, 2-stage cycle potentially decreases COP due to too low pressure ratio (Figure 3.6) [26,136]. In addition, 2-stage cycles are more complex and expensive, thus should be used only when there is a space for performance improvement [17,104].

The suggested practical 1-stage cycle for HCs, HCFOs and HFOs is shown in Figure 7.5 for T_{lift} from 20 to 60 °C. The cycle is the basic cycle equipped with IHX that was gone through already in detail in touch with Figure 6.5. Briefly, the improvement when compared to the basic cycle is that the liquid refrigerant is subcooled more with IHX (3→4) from which the heat is transferred to superheat the vapor before the compressor (6→1) and thus, this increases the useful evaporation enthalpy in the evaporator and the useful desuperheating before the condensation. The usage of IHX allows other refrigerants to behave like R717 with respect to the high amount of the desuperheating and its benefits. Both COP and VHC of HC, HCFO and HFO refrigerants from Table 7.1 increase more or less when IHX is added to the basic cycle depending on the refrigerant and IHX effectiveness (IHX heat transfer area, i.e. ΔT_{pinch}) such that typically COP increases from 10 to 20 % along with VHC increases from 10 to even 40 % [12,24,25,26,37,88,93,97,98,100,105]. As VHC increases, smaller compressors could be needed and could compensate the investment cost of IHX. In addition, the expansion valve operation can be improved due to IHX ensures that the refrigerant is 100 % liquid and thus, excessive pressure losses can be avoided [23,44]. Those refrigerants that have the overhanging saturated vapor curve were not considered promising due to too high T_{crit}, and thus a weak performance with the target heat sink temperatures. But, most of the promising refrigerants have the isentropic saturated vapor curve and thus require a moderate minSH that could be provided with the evaporator. However, as T_{evap} decreases when providing the SH with the evaporator, both COP and VHC decreases as well. Thus, the use of IHX is beneficial to minimize the SH in the evaporator and to provide all the required SH with IHX to maximize T_{evap} and to decrease T_{lift} and thus to improve both COP and VHC. Practical evidence from the usage of IHX gives that it is applied in several of the current commercialized HTHPs along with IHX is considered to give the best compromise between the cycle complexity and the performance [17,107]. Note that the superheating in IHX should not be too high due to the compressor discharge temperature could be a too high as for high T_{cond} and as for some of the promising HTHP refrigerants [25,26,98,100]. Especially, R1234ze(Z) can experience relatively high discharge temperatures from HC, HCFO and HFO refrigerants and thus indicates that its T_{lift} should potentially be restricted up to 50 °C [98]. In addition to the sizing of the IHX heat transfer area, a dynamic refrigerant flow control is possible to adjust the IHX effectiveness according to the compressor discharge temperature [25]. These actions can allow that R1234ze(Z) can be also used for T_{lift} up to 60 °C for a 1-stage compression along with to ensure that the other refrigerants do not experience too high discharge temperatures, too. When T_{lift} is around 50 °C and supplying heat sink of 100 °C with R1234ze(Z), the compressor discharge temperature could be maximally around 120 °C without IHX [88,89]. Thus, there should be space to utilize IHX without increasing the disharge temperature too much.

The suggested practical 2-stage cycle for HCs, HCFOs and HFOs is shown in Figure 7.6 for T_{lift} from 60 to 70 °C along with from 50 to 70 °C for R1234ze(Z) [98]. The cycle combines the 2-stage R717 cycle and the 1-stage HC, HCFO and HFO cycle. Since T_{lift} is enough high, the performance improvements due to the 2-stage compression are clear along with IHX can be justified due to it is beneficial for these refrigerants [26]. However, none of the current commercialized HTHPs do not use IHX when 2-stage cycles are applied, but there is at least one HTHP that utilizes the flash tank in a 2-stage cycle with HFO refrigerant [17]. As for this 2-stage cycle, the compressors can be two separate compressors or a one 2-stage compound since those compound compressors are available for these refrigerants. Moreover, it can be that the IHX effectiveness can be even relatively high due to with the maximum suggested T_{lift} of 70 °C allows that the compressor discharge temperature could potentially be well below 180 °C for heat sinks up to 100 °C except for R1234ze(Z) for which the discharge temperature could be around 180 °C in this condition [98].

As a final cycle configuration for HCs, HCFOs and HFOs, a 2-stage cascade cycle as shown in Figure 7.7 is suggested for very high T_{lift} from 70 °C to even up to 120 °C [28,50,98,100]. The cascade cycle combines two individual 1-stage IHX cycles by using a cascade HX (CAS) that acts as the condenser for the low temperature cycle and as the evaporator for the high temperature cycle. The advantage is that both the low and the high temperature cycles can be optimized with respect to the refrigerants [17,30,31,100]. Thus, the refrigerants from Table 7.1 that have a relatively low T_{crit} (e.g. R1234ze(E)), R290 and R1270) should be used in the low temperature cycle since those perform better in the lower temperatures along with the refrigerants that have a high T_{crit} (e.g. R1224yd(Z) and R1234ze(Z)) should be used in the high temperature cycle. Thus, both the COP and the VHC can be optimized for both the low and the high temperature cycles leading potentially to smaller compressor sizes and thus costs. A drawback is that the CAS needs a certain ΔT_{pinch} in practice that decreases the performance due to slightly increased T_{lift} for both the low and the high temperature cycles and thus leads to that this configuration is not suggested for T_{lift} that are lower than 70 °C but potentially performs better than the 2-stage flash tank cycle for T_{lift} of 70 °C and higher [98]. The compressor discharge temperature issues are potentially in a better control with this cascade cycle when compared to the flash tank cycle due to the low temperature cycle unlikely experiences too high discharge temperatures with T_{lift} up to 60 °C combined with its relatively low T_{cond}. Moreover, the high temperature cycle operates just like the 1-stage IHX cycle and combined with the maximum target heat sink of 100 °C, the T_{lift} of 60 °C for this high temperature cycle can be achieved e.g. with R1224yd(Z) and R1233zd(E) so that the compressor discharge temperature could be below 180 °C along with R1234ze(Z) could be used at least for T_{lift} up to 50 °C to limit the discharge temperature to be below 180 °C [98]. For instance, with e.g. heat sources that cool from 20 to 15 °C in the evaporator along with by assuming ΔT_{pinch} of 5 °C as for all the HXs and the low temperature cycle provides T_{lift} of 60 °C, this could mean that the high temperature cycle could need to evaporate at around 65 °C and thus, the required T_{lift} from the high temperature cycle to achieve heat sink of 100 °C could be around 35…40 °C that should not be an issue from a point of view the discharge temperatures even with high IHX effectivenesses. Practical evidence gives that at least some of the currently commercialized HTHPs apply this cascade configuration without IHX [17,117].

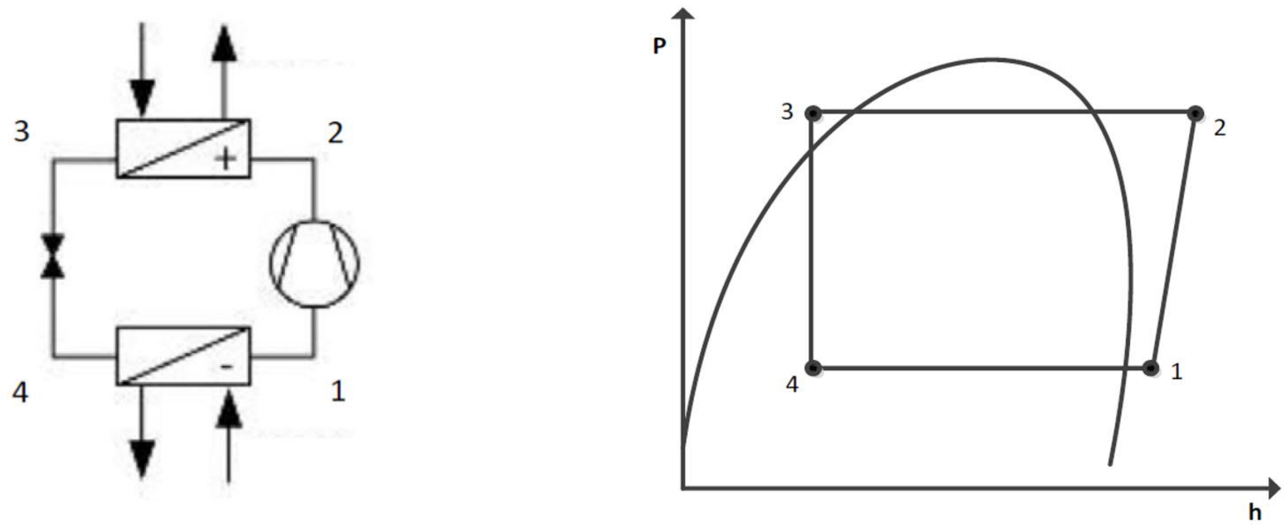

Figure 7.3. Practical R717 cycle configuration for moderate T_{lift} (modified from [17,104]).

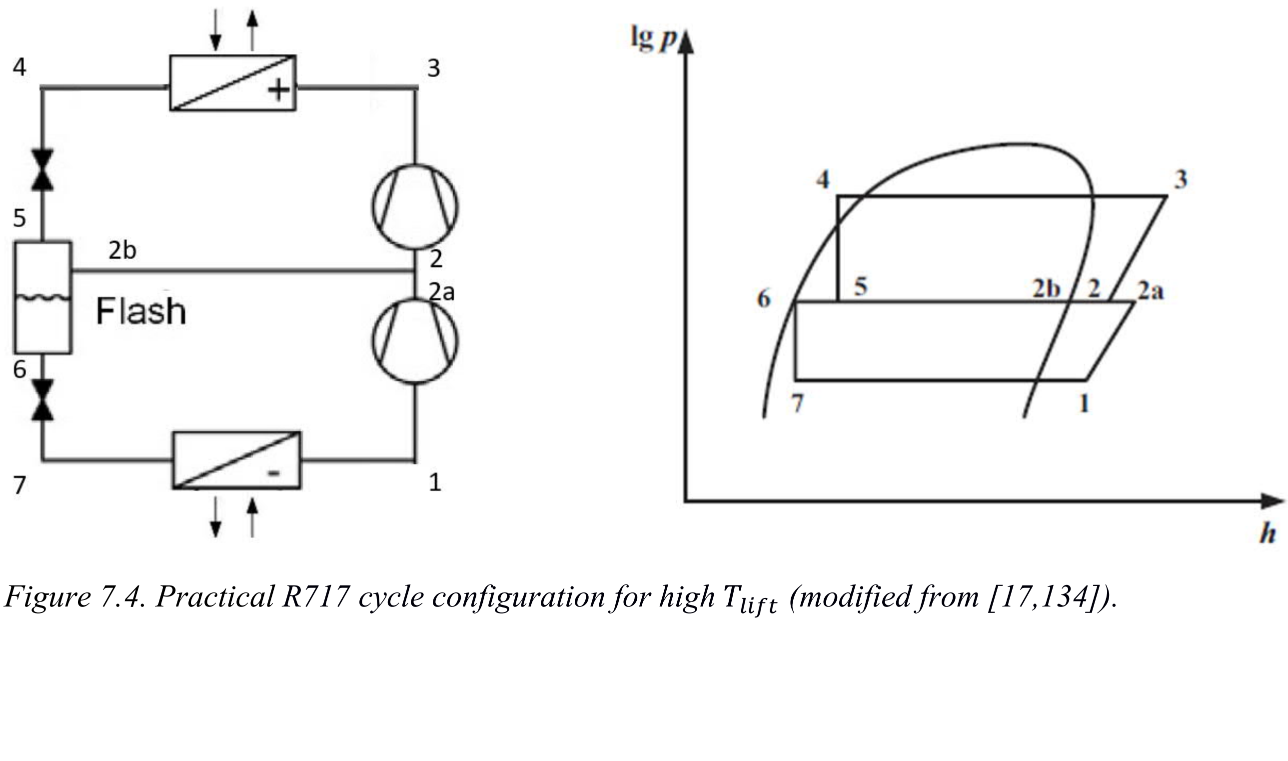

Figure 7.4. Practical R717 cycle configuration for high T_{lift} (modified from [17,134]).

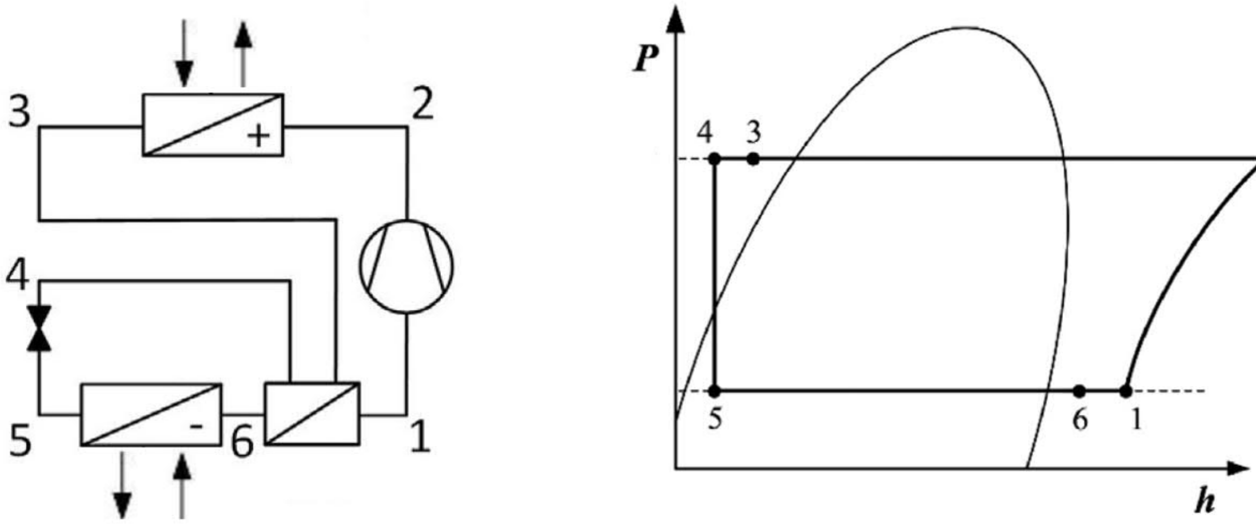

Figure 7.5. Practical HC/HCFO/HFO cycle configuration for moderate T_{lift} [17,26].

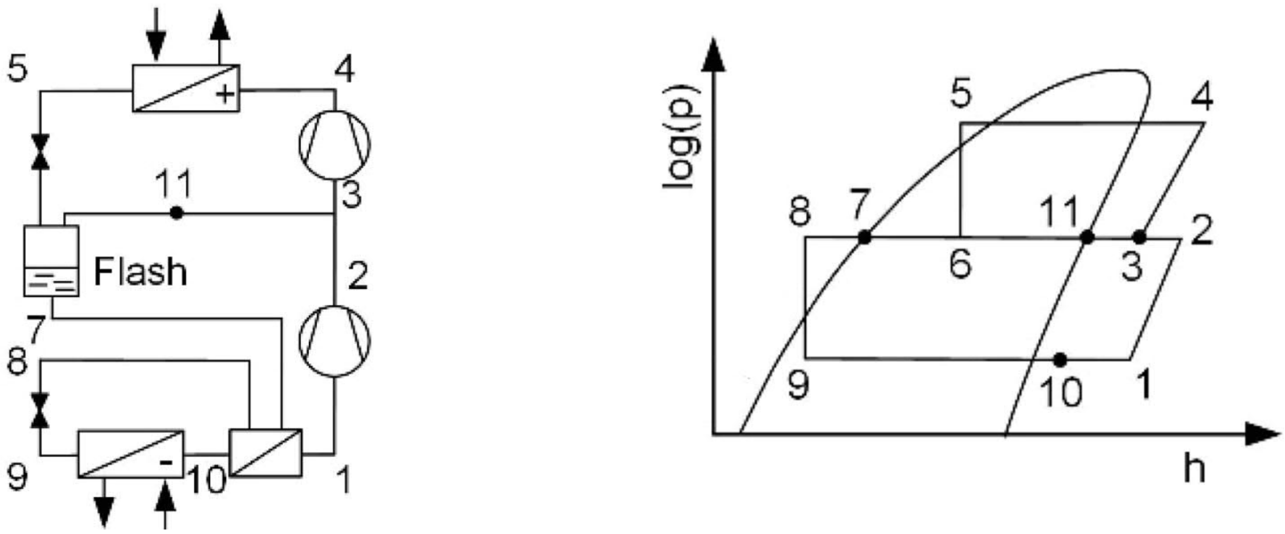

Figure 7.6. Practical HC/HCFO/HFO cycle configuration for high T_{lift} [98].

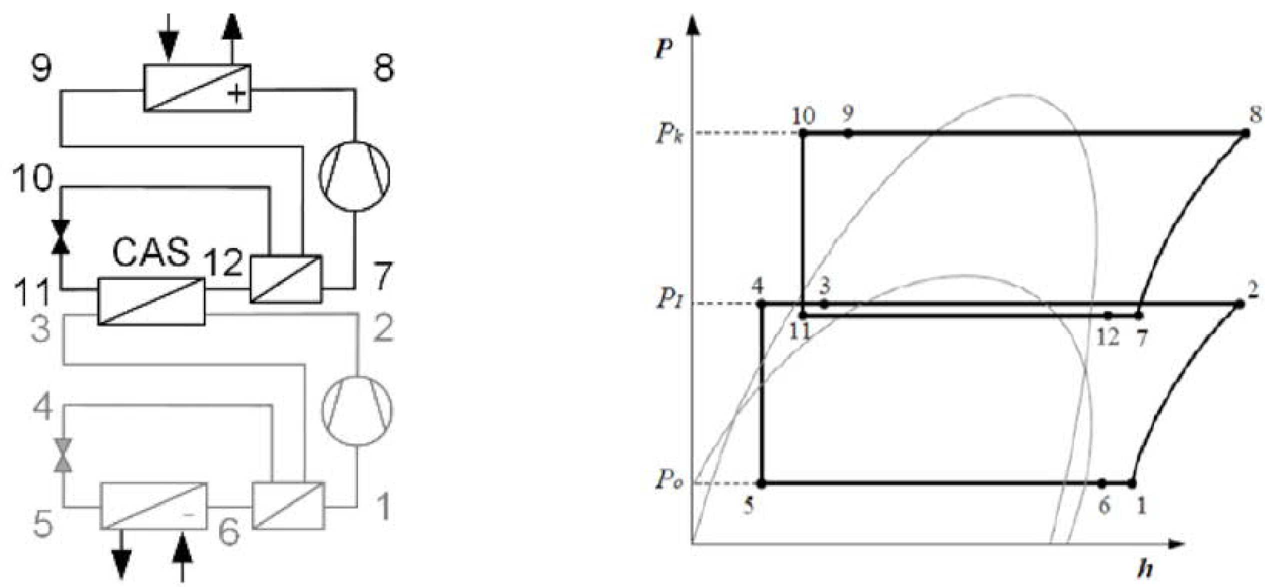

Figure 7.7. Practical HC/HCFO/HFO cycle configuration for very high T_{lift} [98,100].

8 Profitability

Despite HTHPs can improve the energy efficiency and greatly reduce environmental impacts, there are barriers that decelerate the deployment of HTHPs, at least currently. First of all, a payback period requirement is generally few years in the industry which is challenging for HTHPs. Currently, alternative fuel prices, e.g. natural gas or the biomass, are relatively cheap when compared to the electricity causing more challenges for HTHPs to be economically profitable. The simultaneity of the heat demand and the waste heat generation can be weak. Furthermore, there have been identified a general lack of both awareness and knowledge related to the integration of HTHPs to industrial processes. [17,18] The economical profitability seems to be especially a challenge for HTHPs currently. This section estimates the profitability of HTHPs from a point of view the energy economics, the friendliness to the environment (emissions) and the energy efficiency. As a base for the estimation, a case study was implemented. Furthermore, the results were analyzed by using a sensitivity analysis to see how the profitability is dependent on relevant parameters along with where the limits for the profitability could be.

8.1 Case description

The case study is related to a real industrial site in Finland. The client of this thesis organized the necessary information from the industrial actor for this profitability estimation. The site has several processes that require heat at 90 °C and a such waste heat that should be cooled to 40 °C. If the HTHP could be integrated within the industrial site, its required heating capacity could be 500 kW within an annual operating time of 6000 h/a. The waste heat generation is steady and is enough by its capacity. Currently, the heat demand is covered with DH. Figure 8.1 describes the case.

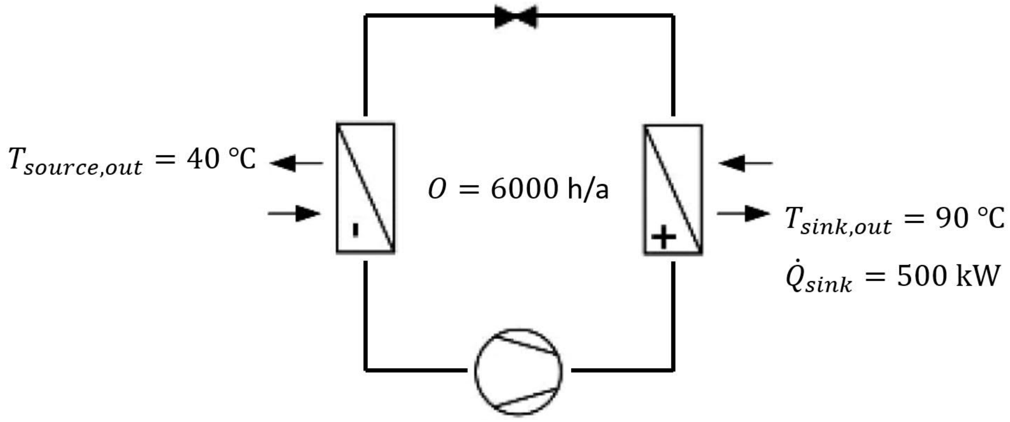

Figure 8.1. Description of the case.

To estimate the profitability of the HTHP, the refrigerant and the cycle configuration must be selected. The investment costs depend on the selected refrigerant [91]. In addition, the achievable performance (COP) of the HTHP depends on the selected refrigerant as shown in section 7.1.2. To estimate the influence of the refrigerant selection on the profitability, the refrigerants R600a, R717 and R1224yd(Z) were selected for this case study. These refrigerants were selected since these could reach a promising performance within the conditions of the case study, see Figure 7.1 for a relative performance. In addition, these refrigerants have varying properties: R1224yd(Z) is a new synthetic refrigerant that is not flammable or toxic, R717 is a natural refrigerant that is both flammable and toxic along with R600a is a natural refrigerant that is not toxic but is highly flammable.

The reference [88] that was utilized for the refrigerant comparison in section 7.1.2 and in section 7.2 is further utilized in this case study. In [88], the T_{lift} was around 60 °C as for the bell-shaped R717, whereas the T_{lift} was around 70 °C for the isentropic R600a and R1224yd(Z). If considering that the conditions could be similar in this case study when compared to [88], the T_{lift} could be around 50 °C as for R717 along with around 60 °C as for R600a and R1224yd(Z) since $T_{sink,out}$ is the same in this case along with $T_{source,out}$ is around 10 °C higher since $T_{source,out}$ was 31 °C in [88]. In the light of the potentially required T_{lift}, 1-stage cycle configurations were selected for the refrigerants since it was concluded in section 7.3 that 1-stage cycles could be feasible for T_{lift} at least up to 60 °C. The investigated refrigerants and the cycle configurations are summarized below. Thus, five cycle configurations were selected from which R600a and R1224yd(Z) were investigated for both the basic and the IHX cycles to further estimate what could be the influence of the IHX on the profitability.

- R717: basic cycle (Figure 7.3)
- R600a: basic cycle (Figure 7.3)
- R600a: basic cycle with IHX (Figure 7.5)
- R1224yd(Z): basic cycle (Figure 7.3)
- R1224yd(Z): basic cycle with IHX (Figure 7.5)

8.2 Methodology

This section goes through the methodology that was applied to estimate the profitability of HTHPs from a point of view the energy economics, the emissions and the energy efficiency. The profitability indicators were kept short and sweet since the purpose was to estimate the profitability in a general level. As for the economics, the profitability of HTHPs is commonly defined by setting an upper limit for the simple payback period (SPP) that is commonly in the industry from 2 to 5 years currently [17,18,20,66,73]. The SPP is simply the total HTHP investment costs I divided by the annual savings S when comparing a HTHP to an alternative heat generator, Equation 11 [66,72,90]. More precisely, I is a difference in the investment cost between two options [137]. So, if replacing an existing heat generator with a HTHP, I is simply the HTHP investment cost. As for the SPP, the potential annual savings can be calculated like shown in Equation 12 [90] as the difference between the operating costs, where the heating capacity $\dot{Q}_{sink}$ divided by the COP is the required electrical power to operate HTHP as shown in Equation 1. As for the total HTHP investment costs, there is a significant variation across cases since it has been estimated and experienced to be from around 200 to 1000 € per installed kW of HTHP heating capacity [12,28,66,72,74,75,84,88,91]. However, there are other more accurate methodologies that consider e.g. the HTHP lifetime, a potential interest rate (time value of money) and changes in the energy prices in time, but there are significant uncertainties [23,137]. Since the required SPP is a relatively low, the influence of the time value of money is likely limited and thus the SPP can estimate the profitability without a significant error [137].

$$SPP = \frac{I}{S} \tag{11}$$

$$S = \left(\frac{\dot{Q}_{sink}}{\eta_{alt}} \cdot C_{alt} - \frac{\dot{Q}_{sink}}{COP} \cdot C_{el}\right) \cdot O \tag{12}$$

As for the emissions, specific CO_2-emissions of supplied heat were compared according to Equation 13 obeying principles in [137]. Note that e.g. DH network or electricity grid transmission losses were not considered. The emissions for an alternative heat generator were defined according to its primary energy CO_2-emission coefficient and the energy conversion efficiency (e.g. boiler efficiency) along with the emissions for a HTHP were defined according to the CO_2-emission coefficient for electricity and the COP. [10] As for Finland, current emission coefficients in 2020 were used to estimate the emissions for fossil fuels, DH, a direct electricity and the electricity-driven HTHP [10]. In addition, Germany, Norway and EU-average were included in the estimation of the HTHP emissions according to coefficients from [28]. Note that $TEWI$ should be in principle used for the HTHP emissions. However, the influence of the refrigerant leakages is negligible since only the low GWP refrigerants were considered [22]. Thus, only the electricity consumption to operate HTHP was considered.

$$\epsilon = \frac{\beta_{alt}}{\eta_{alt}} = \frac{\beta_{el}}{COP} \tag{13}$$

As for the energy efficiency, a coefficient of system performance (COSP) concept was used according to [92], Equation 14. Currently, the electricity has a certain conversion efficiency from a primary energy (e.g. fossil fuel and biomass) to the electricity (secondary energy) since a certain proportion of the primary energy is used with varying conversion efficiencies. The conversion efficiency could potentially be up to 100 % or close to that if only renewable energy technologies are used to generate electricity. Conservatively, the conversion efficiency has been estimated to be 0.4 due to a combustion-based electricity generation. [92] Due to the uncertainty in the conversion efficiency, it was varied between 0.4…1 to show the effects of the transition from the combustion-based electricity generation to renewable technologies. As for the alternative heat generation methods, the energy efficiency is simply the conversion efficiency of a certain alternative heat generator since it stands for the ratio of the useful thermal energy output to the primary energy input, just like the COSP.

$$COSP = \eta_{el} \cdot COP \tag{14}$$

As shown, the COP is needed for all the three ways to estimate the profitability of the HTHP. There are sophisticated ways to estimate the COP that in a nutshell utilize a selected simulation software where the refrigerant properties are included along with the energy conservation is applied within HXs with the aid of the temperature-heat-diagram and a certain ΔT_{pinch} [17,89,100]. The Carnot efficiencies of current commercialized HTHPs are measured to be roughly between 0.3…0.7 [12,17,25,48] from which the most common is around 0.5 [17]. As for the five selected refrigerant-cycle combinations, the COPs were estimated by using the simulated COPs from [88] and the extrapolated COP of R1224yd(Z) that was explained in touch with section 7.1.2. As for the IHX, it was assumed that it improves the COP of R1224yd(Z) as much as R1233zd(E) according to simulations from [37]. Firstly, the Carnot efficiencies were calculated for the conditions in [88] according to Equation 3 and Equation 5. Secondly, the Carnot COP was calculated for this case study according to Equation 3. Thirdly, the calculated Carnot efficiencies from [88] were used in this case study to determine the COPs of the five refrigerant-cycle combinations according to Equation 5 by assuming that the efficiencies could remain the same with the around 10 °C higher $T_{source,out}$ of this study. It was deemed that this simplified way estimates the COP in a sufficient way. In addition, the sophisticated ways to estimate the performance of HTHPs were out of the scope of this thesis.

8.3 Profitability analysis

This section goes through the remaining source information and the remaining parameters. Moreover, the results are shown and discussed. Table 8.1 shows the used COPs of the studied HTHPs in this case study. The COP_{ref} is for the operating conditions in [88]. The used Carnot efficiencies and so COPs were calculated like described in the end of section 8.2. Moreover, the client estimated the total investment costs of the HTHPs including the integration costs in site to the industrial processes like shown in Table 8.1. Despite the R717 has exceptionally high VHC, the R717 system could have some premium cost due to the higher pressures, e.g. the compressor is needed to be suitable. Moreover, R600a has some premium costs due to the safety measures, but the required compressor size is smaller than that of with R1224yd(Z) and so, the investment cost of R600a could be smaller. Like shown, the addition of IHX causes some additional costs, somewhat higher as for R1224yd(Z) due to larger components when compared to R600a.

Table 8.1. Performance of the selected refrigerant-cycle combinations.

		COP_{ref}	η_{car}	COP	I
		[-]	[-]	[-]	[€]
1-stage basic	R717	3.91	0.64	4.65	310000
	R600a	2.79	0.45	3.27	210000
	R1224yd(Z)	3.36	0.55	3.99	240000
1-stage IHX	R600a	3.50	0.57	4.14	222000
	R1224yd(Z)	3.76	0.61	4.43	253000

Table 8.2 shows the remaining source information and parameters that have not been yet mentioned. The conversion efficiency from the DH network to the heating circuit of the industrial site and the CO_2-emission coefficients are up to date according to the Ministry of the Environment of Finland. In addition, a conversion efficiency of a primary-energy-fired heat boiler was considered in the COSP for the DH since the DH is first converted in a DH plant from a primary energy to a DH network. The price of the DH and the electricity are such that the industrial actor pays currently.

Table 8.2. Source information and parameters.

$C_{alt,DH}$	C_{el}	$\beta_{alt,DH}$	β_{el}	$\eta_{alt,DH}$	$\eta_{alt,boiler}$	
[€/kWh]	[€/kWh]	[g_{CO2}/kWh]	[g_{CO2}/kWh]	[-]	[-]	
0.065	0.080	130	121	0.97	0.88	
client	client	[10]	[10]	[138]	[12,16,28,66,88,92,139]	reference

Table 8.3 summarizes the results of this case study. R717 is the most profitable from a point of view the energy efficiency and the emissions along with R600a seem to have the worst profitability. However, R600a is the most profitable with respect to the economics having SPP of around 1.6 years, whereas R717 has the worst economical profitability that is around 30 % higher than that of with R600a. Addition of the IHX seem to be clearly profitable, where the improvement as for R600a is significant but R1224yd(Z) performs still better than R600a when the IHX is added for both. However, the addition of the IHX did not show extra benefits in the economical profitability for the

case at hand. All in all, the case seems to be economically profitable since the SPP could be around 2 years. As for the emissions, the HTHP could be clearly profitable since the emissions could be reduced around 3.6…5.2 times. The energy efficiency of the HTHP could be clearly profitable even with the conservative η_{el} of 0.40 because the COSP of the HTHP could be around 1.5…2.2 times higher than that of DH. Thus, the HTHP seems to be clearly profitable in all the three matters for the case at hand. The use of IHX is suggested as for R600a and R1224yd(Z) due to the profitability with respect to the energy efficiency and the emissions is improved without decreasing the economics.

Table 8.3. Results of the case study.

		SPP	ϵ	$COSP_{40\%}$	$COSP_{70\%}$	$COSP_{100\%}$
		[a]	[g_{CO2}/kWh]	[-]	[-]	[-]
1-stage basic	R717	2.1	26	1.86	3.25	4.65
	R600a	1.6	37	1.31	2.29	3.27
	R1224yd(Z)	1.7	30	1.60	2.80	3.99
1-stage IHX	R600a	1.6	29	1.66	2.90	4.14
	R1224yd(Z)	1.7	27	1.77	3.10	4.43
	DH		134			0.85

The validity of the Carnot efficiencies from Table 8.1 was estimated as follows. As for the basic cycle within the conditions of the case study, T_{lift} could be around 50 °C as for the bell-shaped R717 along with around 60 °C as for the isentropic refrigerants R600a and R1224yd(Z). The behavior for the demanded T_{lift} was checked according to the vapor pressures from the reference for the Carnot efficiencies [88] and the CoolPack software [108]. Moreover, the required temperature differences between T_{evap} and $T_{source,out}$ along with T_{cond} and $T_{sink,out}$ as for R717 and R600a were checked by using the mentioned references. The resulting COPs with potential piston compressors was checked by using GEA RTSelect (R717) software [132] and Frascold Selection (R600a) software [140]. R1224yd(Z) was not available in the software, thus its validity was assumed according to R600a due to similar behavior with respect to the saturated vapor curve. Table 8.4 summarizes the estimation of the validity where the bold Carnot efficiencies are based on the COPs from the software. The Carnot efficiencies (Equations 3 & 5) were calculated to be around 0.68 as for R717 along with around 0.45 as for R600a when subcooling or superheating were not applied like in [88]. As shown in Table 8.4, the temperature differences between T_{evap} and $T_{source,out}$ along with T_{cond} and $T_{sink,out}$ are kept the same as in [88] to keep the Carnot efficiencies comparable. The COPs from the compressor software indicates roughly the performance that could be achievable in practice. All in all, the used Carnot efficiencies in this case study seem to be well valid because they are in a good agreement with those that were estimated with the compressor software.

Table 8.4. Validity of the selected performance.

	T_{cond}	$T_{sink,out}$	T_{evap}	$T_{source,out}$	COP	COP_{car}	η_{car}	reference
	[°C]	[°C]	[°C]	[°C]	[-]	[-]	[-]	
R717	87	90	26	31	3.91	6.16	0.64	[88]
R600a	96	90	26	31	2.79	6.16	0.45	[88]
R717	70	73	20	25	4.87	7.21	**0.68**	GEA [132]
R600a	75	69	15	20	3.11	6.98	**0.45**	Frascold [140]

8.4 Sensitivity analysis

In the previous section, the profitability was estimated with the case-specific values. In addition, the performance of the HTHP was estimated according to available data. However, it is possible that the values can change in the future along with e.g. the estimated performance and the investment costs of the HTHP can be somewhat else. To investigate the influence of the variations in the parameters, a following sensitivity analysis was implemented. The shown procedure is suggested e.g. in [137].

Figure 8.2, 8.3 and 8.4 shows the sensitivity of the economical profitability, the emissions and the energy efficiency respectively. As the base case, the R1224yd(Z) HTHP with IHX like listed in section 8.1…8.3 was used that is at a change in a parameter of 0 % on the horizontal axis. Note that only one parameter is varied at a time while the others were according to section 8.1…8.3. In this analysis, electricity to heat conversion efficiency for an electricity boiler was assumed to be 0.95 [16].

According to Figure 8.2, the economical profitability is the most sensitive with respect to the fuel price (DH) such that the SPP could be over 5 years if the price gets more than 50 % cheaper. In addition, the annual operation hours and Carnot efficiency should be high along with the electricity price, the investment costs and T_{lift} should be enough low to ensure the economical profitability.

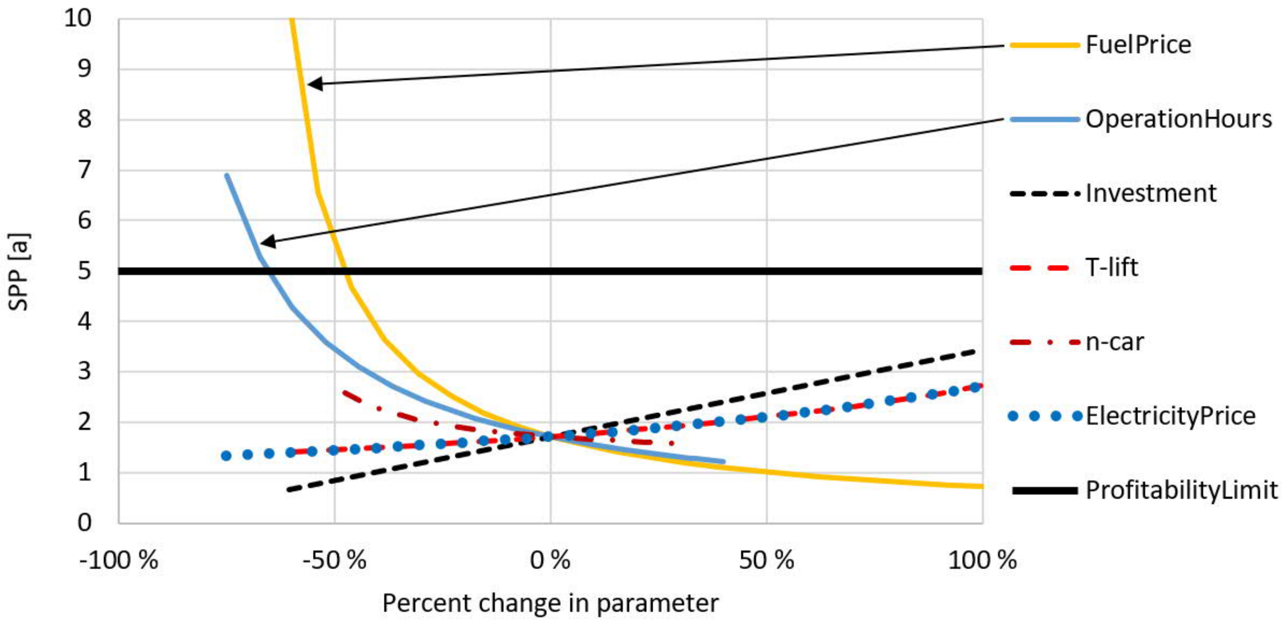

Figure 8.2. Sensitivity of the economical profitability.

As shown in Figure 8.3, the emissions from the heat supply with the HTHP are multiple times lower in several cases when compared to a fossil-fuel fired heat boiler, an electricity boiler and the DH. However, the emissions from the electricity generation vary, at least currently and influences relatively much the emissions of the HTHP operation. If electricity-driven boilers were used in Norway, on average in the EU or in Germany, the emissions from the heat supply could be 23, 291 or 447 g-CO_2/kWh respectively. Norway has exceptionally low emissions from the electricity generation indicating that even the direct use of electricity in heat boilers has relatively low emissions. Even with the electricity generation emissions in Germany and high T_{lift} or low Carnot efficiencies, the HTHP could have less emissions than Finnish fossil fuels and DH in several cases. As for the Finnish case, the HTHP has clearly lower emissions than the alternative heat generation methods.

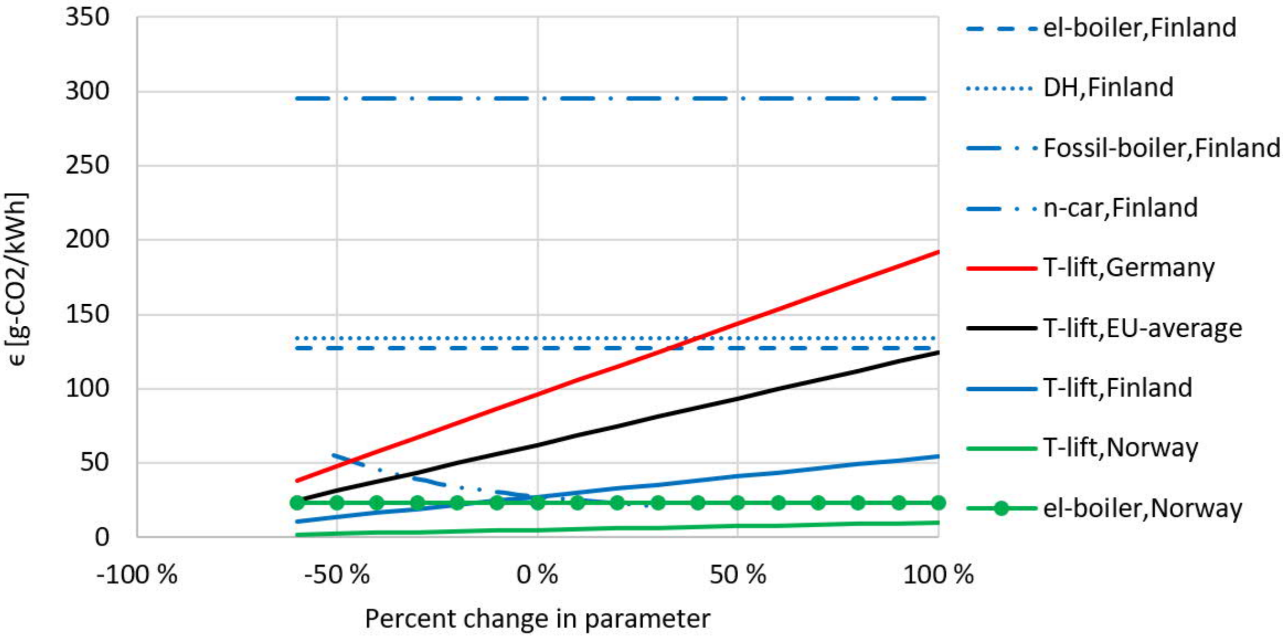

Figure 8.3. Sensitivity of the emissions.

Figure 8.4 shows that the energy efficiency of the HTHP could be higher almost in all of the shown cases, even multiple times higher in several cases, than that of the alternative heat generation methods. The COSP of DH was estimated to be 0.85 like shown in Table 8.3. The influence of the conversion efficiency from the primary energy to the electricity is shown as for the COSP with varying Carnot efficiency and T_{lift} of the HTHP along with as for the electrical boiler. For instance, the fan of the curves with varying T_{lift} shows that the COSP is lower with a lower electricity conversion efficiency. The conversion efficiency of unity (100 %) corresponds the COP. The conversion efficiency should be over 0.4 due to certain proportion of renewable electricity generation currently, but the conversion efficiency depends on e.g. the country due to varying mixes in the electricity generation. In the future, the conversion efficiency can be expected to be clearly higher than 0.4 due to the climate targets.

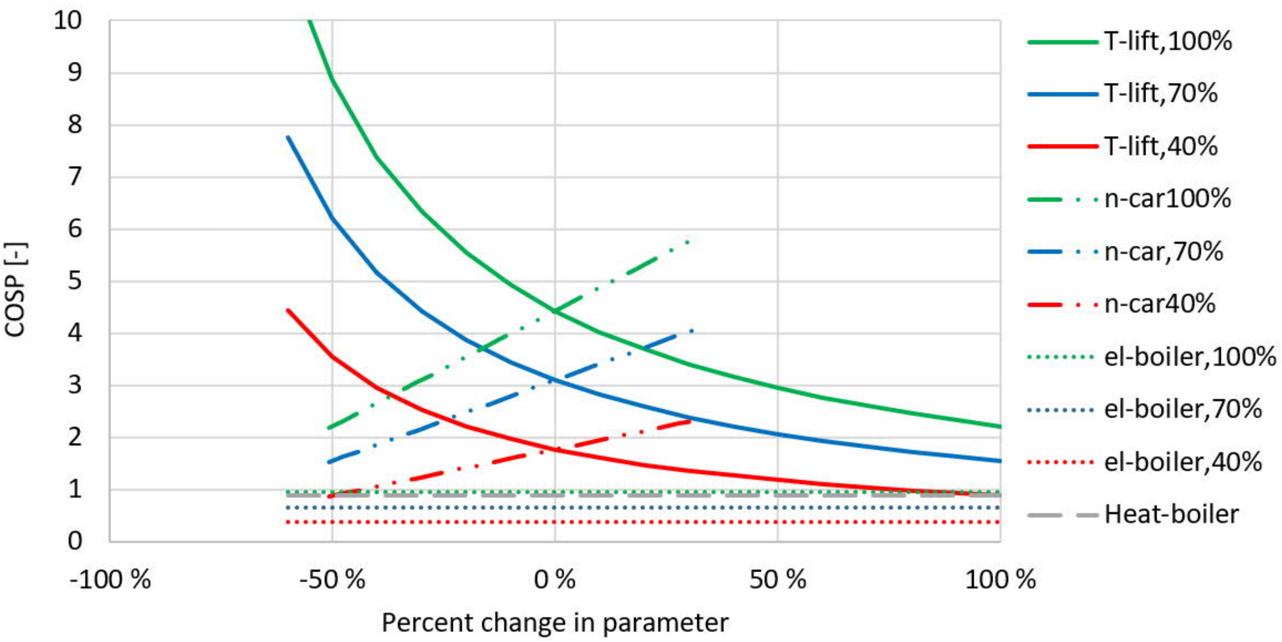

Figure 8.4. Sensitivity of the energy efficiency.

9 Conclusions

All in all, at least 32 commercialized HTHP models are available currently. Several models can produce heat sinks of at least 100 °C. Majority of the models apply either R717, R1234ze(E) or high GWP HFCs R134a and R245fa as the refrigerant. The heating capacities vary considerably from around 10 to 20000 kW from which in general the smaller heating capacities are provided with piston compressors, the intermediate with screw compressors and the large with turbo compressors.

According to the survey, it is evident that there are applications where HTHPs could be applied. When considering the industrial potentials, there is estimated to be e.g. in Finland a waste heat potential of around 16 TWh/a currently that is mostly at temperatures below 100 °C. At the same time, there is a multiple times larger industrial heat demand, around 145 TWh/a in Finland currently, at significantly varying temperatures. According to the processes within industrial sectors (Appendix 3), there should be applications where the demand is at around 100 °C and below that. The estimated low-temperature waste heat flows are technically an attractive source to produce heat sinks of around 100 °C by using HTHPs. In addition to the industrial applications, DH networks seems to be a technically attractive application for HTHPs since the heat demand is minimally at around 75 °C and in practice maximally at around 115 °C e.g. in Finland, currently. The highest demanded temperatures occur only when the outdoor air temperature is very low leading to that clearly a majority part of the total DH demand (around 35 TWh/a in Finland currently) could be technically covered with HTHPs that produce heat sinks up to 100 °C. Furthermore, there seems to be several waste heat sources in the built environment that are mainly at temperatures of 10…60 °C. Moreover, the industrial waste heat can be recycled into a DH network if there are not applications where the waste heat could be utilized. And, the waste heat from the built environment can be recycled into the industry. Both the waste heat generation and the heat demand should be simultaneous. Both in the industry and in the DH networks, there are lots of processes that have efficiency improvement potentials. For instance, the simultaneous heating and cooling is a one especially attractive option since the COP could be exceptionally high and at the same time, HTHP can potentially replace both conventional heating and cooling generators.

Both the refrigerant and the compressor were identified to be clearly the most important HTHP subjects. The refrigerants vary significantly by their thermodynamic properties. The properties are more or less suitable for certain temperature levels. At some fixed heat source and sink temperatures, some refrigerants simply perform better than others. At desired temperature levels, the achievable COP and VHC were identified to be the most important considerations when selecting the refrigerant. Furthermore, it is currently evident that the refrigerant cannot be ODS. The GWP should be as low as possible, preferably 0…10. There is both an experimental and a practical evidence on that the VHC should be at least 1000…1500 kJ/m^3 to avoid it from decreasing the practical COP. As for both COP and VHC, T_{crit} and the refrigerant vapor pressures are especially important refrigerant properties from which higher T_{crit} indicates a higher achievable COP at an optimized T_{cond} that is generally 20…60 °C below T_{crit}. As for subcritical cycles, T_{crit} must be at least 10…15 °C higher than that of T_{cond} to achieve a reasonably high COP. Moreover, higher vapor pressures indicate a higher VHC meaning that smaller compressors could be required for a certain heating capacity.

As for the compressors, the technology has some limitations. Firstly, the suction and discharge pressures are limited indicating limitations for both T_{evap} and T_{cond} respectively depending on the refrigerant. The compressors that are suitable for the target heating capacities of around 50…1000 kW, are limited to condensation pressures up to around 60 bar as for R717 and generally up to around

30 bar as for the HC, HFO and HCFO refrigerants. Secondly, the compressor discharge temperature shall not be higher than 140…180 °C. Thus, both the compressor and the refrigerant should be selected together. The refrigerant vapor pressures determine the compressors that could be suitable according to the pressures or vice versa. There is a significant variation in the degree of superheat that a refrigerant experience during the compression. This superheat shall not be too high to avoid a decomposition of the compressor lubrication oil and to avoid excessive compressor wear rate. Moreover, increasing T_{crit} has been found to increase the refrigerant molar mass that in general leads to an increasing minimum required refrigerant superheat in the compressor suction to avoid the wet compression. Since T_{crit} should be enough high for HTHPs, several suitable refrigerants require some minimum superheat that is suggested to be implemented with the IHX. Moreover, the IHX is suggested for the suitable refrigerants (excluding R717) because both the COP and the VHC can increase at least up to 20 % when compared to a cycle configuration that does not utilize the IHX.

As for subcritical HTHPs, there are several environmentally friendly refrigerants that could be suitable at least for heat sinks up to 100 °C. Both the achievable COP and VHC have been studied in several research papers that are in a good general agreement. R717 seems to be an exceptionally attractive since it could have both the highest COP and VHC. However, R717 is both toxic and flammable and thus it can be that it is not accepted in all cases. Some synthetic refrigerants, such as R1234ze(Z) and R1224yd(Z) are promising alternatives when the toxicity and the flammability are an issue. Moreover, if R717 is not accepted, R1234ze(E), R600 and R600a are promising alternatives for heat sinks of at least 80…90 °C due to those could achieve higher VHC than R1234ze(Z) and R1224yd(Z), fundamentally due to a higher vapor pressures at those temperatures. Moreover, R290 could be suitable for heat sinks up to around 80 °C having a relatively high VHC. However, the high flammability of HCs can restrict their use where the required safety measures can be an issue. All in all, the VHC should be enough high to avoid a decrease in the practical COP like shown with R1336mzz(Z) whose COP and VHC have been experimentally measured to be relatively low for heat sinks at least below 110 °C [97]. This means that T_{crit} should not in principle be too high for target temperature levels because a too high T_{crit} indicate low vapor pressures and thus a too low VHC. All in all, a more accurate combined cost and emission analysis is strongly suggested find out the optimal options with respect to certain preferences within certain target heat source and sink temperatures.

According to the achievable pressure levels and the maximum compressor discharge temperature, the current commercialized compressors could be suitable for the promising HTHP refrigerants. However, there are some mismatches with the compressor displacements and the pressure levels. The displacements could be handled efficiently with a compressor speed control and by using parallel installed compressors. The pressures are partly an issue for R717, where a compressor development is required. However, one R717 compressor can achieve a maximum discharge pressure that corresponds T_{cond} of around 98 °C meaning that heat sinks of 100 °C or somewhat less could be achieved depending on the proportion of the useful desuperheating before the condensation. As for HTHP heating capacities up to at least 1000 kW, the piston compressor seems to be the most promising compressor technology due to the displacements and pressures correspond well the demand along with the pistons can maintain a high efficiency within varying operation conditions.

According to the survey, the cycle configuration should be selected on basis T_{lift} and the refrigerant. In general, T_{lift} that are up to 60 °C, 1-stage cycles should be used, whereas higher T_{lift} could be technically feasible with 2-stage cycles for T_{lift} up to even 120 °C. As mentioned earlier, IHX should be used because it can increase both the COP and the VHC at least up to 20 %. But, IHX should not be applied for R717 because IHX could even decrease the performance. It should be noticed that the usage of IHX increases the compressor discharge temperature. Thus, the IHX heat transfer effectiveness can require an active control to keep the discharge temperature below 140…180 °C. In addition, the exceptionally high discharge temperature of R717 potentially limit its highest T_{lift} up to around 95 °C when implementing the cycle configuration in two compression stages.

As for the Finnish case study, the operation of the investigated five HTHPs could currently emit on average 4.5 times less equivalent CO_2-emissions and be around 1.5…5.5 times more energy efficient when compared to the average DH. High T_{lift}, low Carnot efficiency (COP) and low efficiency of the electricity generation (η_{el}) increases the emissions and decreases the energy efficiency of HTHPs. Currently, the emissions of the electricity generation vary significantly across countries. In the future, the emissions of the electricity generation can decrease because it is expected that the penetration of renewable energy technologies increases. The VRE generation is expected to increase in the future and thus the electricity price can intermittently decrease when the VRE penetration is high. According to the climate targets, it is evident that the deployment of the renewable technologies must increase. Thus, the friendliness to the environment and the energy efficiency of HTHPs can be expected to increase. And, the overall profitability can benefit from the intermittently lower electricity prices. As for the case study, the HTHP was estimated to be economically profitable as well because the SPP was around 2 years that is within the typical limits of around 2…5 years. According to the sensitivity analysis, the economical profitability of the HTHP is highly sensitive with respect to variations in the source information, where especially both the DH price and the annual process operation hours should be enough high in the case study. Thus, the uncertainty in a single parameter should be clearly within a range where the SPP is less than its upper defined limit. In the light of the case study, the integration of the HTHP to the industrial site is clearly profitable. The case study showed that the choice of the refrigerant leads to a compromise between the performance and the economics. Note that the SPP does not show what HTHP configuration is economically the most profitable during the whole lifecycle, that can be e.g. 20 years. To compare in a more holistic way, the whole lifecycle of all the options must be analyzed, by using e.g. the net present value method, like shown e.g. in [137].

The industrial heat demand that could be technically covered with HTHPs could even double if the maximum achievable heat sink is increased from 100 to 150 °C. The mentioned heat sink temperature gap is technically possible to reach with the technologies covered in this thesis. However, when approaching the heat sinks of around 150 °C, the compressor discharge temperature is more likely to be near the higher limit when comparing to e.g. the heat sinks of 100 °C. Moreover, the refrigerant choice should be suitable for the higher temperatures. From the refrigerants discussed in this thesis, R1336mzz(Z) is one suitable option for heat sinks of at least 150 °C since it is applied already in a commercialized HTHP generating heat sinks up to 165 °C and its vapor pressure is not excessively high in those temperatures (less than 25 bar) along with its T_{crit} is a sufficiently high for a reasonable COP and its saturated vapor curve is the overhanging one meaning that its compressor discharge temperatures are not excessively high. In the big picture, just by changing the refrigerant along with by adjusting the compressor lubrication oil and displacements, the HTHP application potential could be enlarged indicating increased marketing and energy efficiency improvement potentials.

As mentioned in section 1.1, the mixtures were not considered in this thesis. Like discussed in section 3, there could be a significant improvement potential in the HTHP performance because the zeotropic mixtures are expected to behave according to Equation 4 that can achieve better performance than Equation 3 that is in principle valid for pure and azeotropic refrigerants. However, the zeotropic mixtures could require more case-specific investigations to match the temperature glides that could increase investment costs. All in all, the potential of the zeotropic mixtures should kept in the mind.

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Appendices

Appendix 1. Summary: significant findings and references. 3 pages.

Appendix 2. Energy potentials, temperatures and references. 2 pages.

Appendix 3. Demanded temperatures in industrial processes. 1 page.

Appendix 4. Compressor displacements and pressures. 2 pages.

Appendix 1. Summary: significant findings and references

subject	summary	reference
The most significant review on HTHPs: Markets, state of the art, research status, refrigerants and application potentials.	Internationally, over 20 commercialized HTHP models are available. Refrigerant selection criteria for HTHPs discussed extensively. The higher refrigerant Tcrit, the higher COP could be achieved at optimized Tcond that could be around 40…60 °C less than Tcrit. A significant amount of industrial process heat demand is identified between 100…150 °C within food, paper&pulp and chemical industries.	[17]
A theoretical study: technological restrictions for R717 in HTHP applications	R717 has a relatively high performance within compressor technology limits and has exceptionally high compressor discharge temperature. 2-stage cycle enlarges R717 working domain due to lower discharge temperature.	[92]
A theoretical study: basic 1-stage cycle - comparing refrigerants suitable for HTHPs. Heat supply as for heat sinks of around 100 °C studied.	R717 was discussed to have high performance, but technology restricts it for heat sinks up to around 90…100 °C. R744 is not suitable for high temperature heat sources and small temperature glide heat sinks. High GWP refrigerants are not sustainable. For the studied operation points, HCs, HCFOs and HFOs were concluded to be suitable.	[89]
Experiments: R290-R600 cascade HTHP, heat sink up to 115 °C.	With T-lift of 100 °C, COP for heating was measured to be 2.1. Potentially, the compressor discharge temperature could be allowed to be higher when solving issues with the compressor manufacturer.	[28]
A theoretical study: comparison of cycle configurations.	1-stage cycle with IHX is recommended for T-lifts up to around 60 °C and 2-stage cycle with IHX for higher T-lifts. However, with low T-lifts, low pressure ratios of 2-stage cycles cause low isentropic compressor efficiencies and thus result in worse performance than a 1-stage cycle.	[26]
A theoretical study: cycle comparison, high T-lifts	73 °C T-lift: 2-stage compression with intermediate refrigerant vapor injection implemented with a flash tank was found to be cost-optimal.	[104]
A theoretical study: refrigerant comparison, 90 °C heat sink	R717 had both the highest COP and VHC from sustainable and suitable refrigerants for HTHPs. R717 require relatively high pressures which restrict its use in higher temperatures than 90…100 °C.	[88]

Experiments: R1224zd(Z), R1233zd(E) and R1336mzz(Z)	Integration of IHX into the basic 1-stage cycle increased around 15 % COP. With heat sinks below 110 °C, both R1224yd(Z) and R1233zd(E) have higher COP and significantly higher VHC than R1336mzz(Z). With heat sinks below 110 °C, R1224yd(Z) had around 10 % better VHC than R1233zd(E) and COP was the same within the measurement errors.	[97]
A theoretical study: R1224zd(Z), R1233zd(E), R1234ze(Z), R1336mzz(Z) performance.	R1336mzz(Z) has very low VHC for around 100 °C heat sinks. With T-lifts of 70 °C and higher, 2-stage cascade cycle with IHX in both stages showed better COP than other cycles. Lower than 50...60 °C T-lifts could be feasible with 1-stage IHX cycle. R1234ze(Z) showed clearly the highest compressor discharge temperatures. The discharge temperature can be reduced by using 2-stage cycles.	[98]
A theoretical study: R1336mzz(Z), R1233zd(E) and R1224yd(Z), minimum required superheat, IHX benefits	Minimum required refrigerant superheat to ensure dry compression depends mainly on refrigerant thermodynamic properties, R1336mzz(Z) require relatively high. When compared to basic 1-stage cycle, IHX could increase both COP and VHC up to 40 % depending on IHX effectiveness.	[37]
Experiments: influence of VHC on practical COP. Theoretical: IHX benefits	Small VHC decreases practical COP due to larger refrigerant flow rates cause larger heat losses and increase compressor electricity consumption. VHC should be at least around 1500 kJ/m3 to avoid the performance reduction. The IHX is recommended to implement the minimum superheat to prevent the wet compression.	[25]
A theoretical study: HTHP optimization with IHX	As for both R717 and R718, both COP and VHC decreases if IHX is applied. In general, IHX increases both COP and VHC for other refrigerants than R717 and R718. R717 has very high VHC.	[12]
A theoretical study: ODP of HCFOs	ODP of HCFOs is negligible due to their atmospheric lifetime is low.	[122]
Survey: energy potential of industrial waste heat	HTHPs are considered as one of the promising technologies to recycle waste heat in the industry. Discussion about Finnish waste heat quantities.	[5]
A theoretical study: industrial waste heat: HTHPs	The most promising industrial sectors: food, paper&pulp along with chemical due to those have relative much heat demand below 150 °C.	[66]

Survey: industrial waste heat potential in Finland	Estimated industrial waste heat energy quantities and experienced temperature levels in 2010.	[68]
(2nd) Book of presentations related to HTHPs: Potential, case studies, current development and trends related to HTHPs	Increasing Variable Renewable Electricity (VRE) increases HTHP potential. Evidence on heat demand by industry and demanded temperature levels in industries and processes. VRE can be stored by using HTHP and thermal energy storages. Industries and processes with simultaneous cooling and heating demand are promising.	[50]
(1st) Book of presentations related to HTHPs: Potential, case studies, current development and trends related to HTHPs	Especially, food, paper&pulp and chemical industries have significant heat demand at temperature levels below 150 °C. Increasing penetration of Variable Renewable Electricity increases HTHP potential. Evidence on heat demand by industry and demanded temperature levels in industries and processes. Industrial drying has significant energy efficiency improvement potentials that could be achieved by using HTHPs.	[48]

Appendix 2. Energy potentials, temperatures and references

Table 1. Industrial energy potentials: data.

industry	T<55 °C waste energy [TWh/a]	T>55 °C waste energy [TWh/a]	total energy use [TWh/a]	lower waste $T_{source,in}$ [°C]	upper waste $T_{source,in}$ [°C]	lower demand $T_{sink,out}$ [°C]	upper demand $T_{sink,out}$ [°C]
textile and leather			0.2	50	70	70	190
transport equipment			0.7			60	500
plastic			1.2	23	270	30	380
machinery			2.0	20	160	60	500
non-metallic minerals		0.4	3.6	70	1000	100	1000
food and beverage	1.4	0.4	5.0	22	98	50	300
woodworking	0.8	0.6	6.9	50	135	60	300
coke & oil refining	2.1	1.1	11.8	20	600	50	1000
chemical	0.9	0.1	12.6	30	260	50	600
metal refining	0.8	1.7	20.1	20	700	60	1000
paper and pulp	3.5	2.3	80.3	20	120	40	400

Table 2. Industrial energy potentials: references.

textile and leather			[141]	[12]	[12]	[12, 17, 48, 50]	[12, 17, 48, 50]
transport equipment			[141]			[17, 48, 50]	[48, 50]
plastic			[141]	[142]	[142]	[143]	[143]
machinery			[141]	[48]	[48]	[17, 48, 50]	[48, 50]
non-metallic minerals		[5, 68]	[141]	[68]	[68]	[66]	[68]
food and beverage	[5, 68]	[5, 68]	[141]	[68]	[68]	[12, 17, 21, 50, 66, 144-146]	[12, 17, 50, 60, 144]
woodworking	[5, 68]	[5, 68]	[141]	[74]	[68]	[17, 21, 50, 66]	[17, 50, 66]
coke & oil refining	[5, 68]	[5, 68]	[141]	[68]	[68]	[147]	[148]
chemical	[5, 68]	[5, 68]	[141]	[48, 50, 74]	[48, 50]	[17, 21, 50, 66, 146]	[17, 50, 66]
metal refining	[5, 68]	[5, 68]	[141]	[68]	[68]	[17, 50, 66]	[66]
paper and pulp	[5, 68]	[5, 68]	[141]	[68]	[68]	[12, 17, 21, 50, 66, 146]	[17, 50, 66]

Table 3. District heating energy potentials: data.

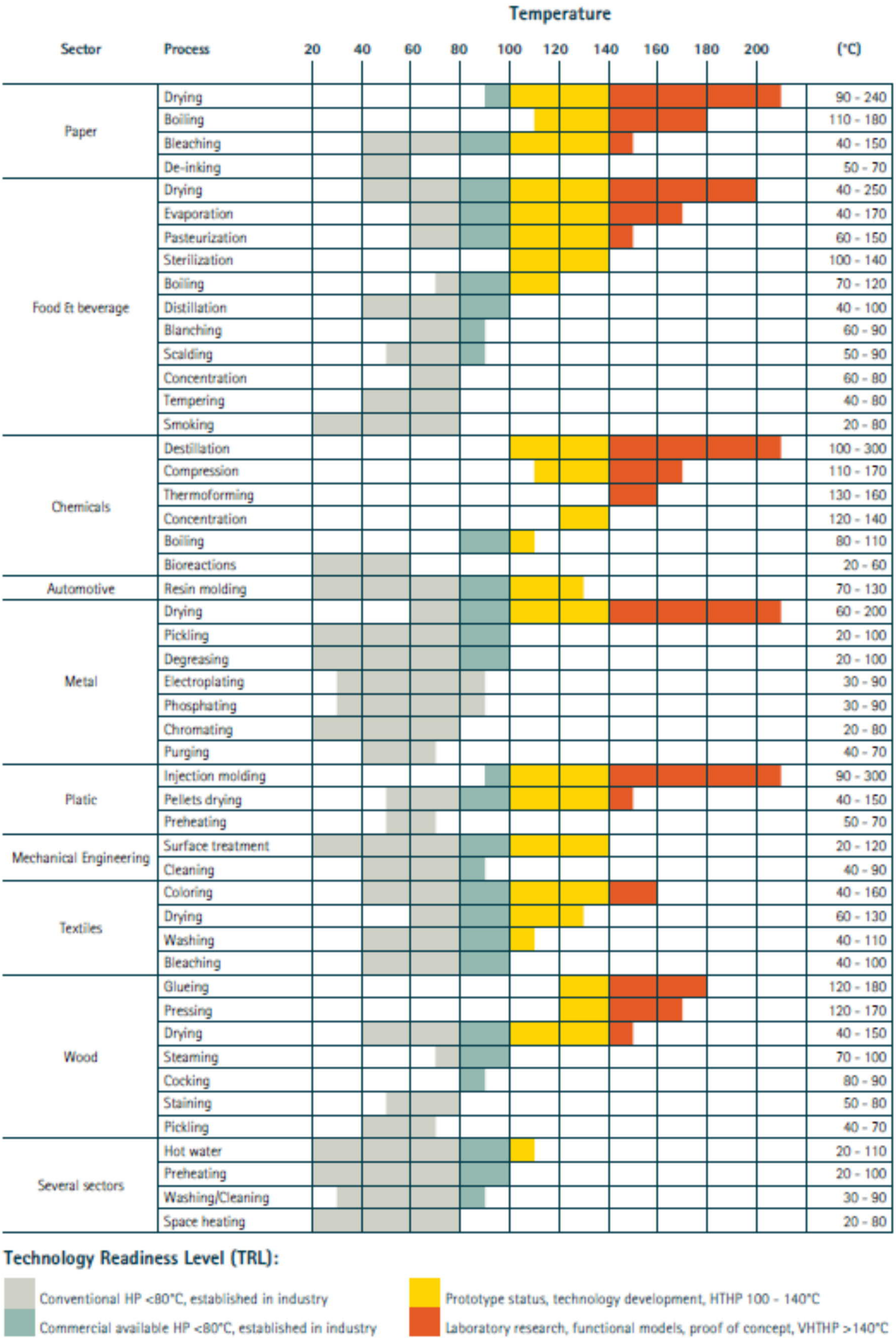
application	T<55 °C waste energy [TWh/a]	T>55 °C waste energy [TWh/a]	total energy use [TWh/a]	lower waste $T_{source,in}$ [°C]	upper waste $T_{source,in}$ [°C]	lower demand $T_{sink,out}$ [°C]	upper demand $T_{sink,out}$ [°C]
DH network			35.4	16	60	75	120
wastewater	1.5			7	30		
datacenter	0.85*			25	60		
CCHP				30	40		
Flue gas				50	60		
refrigeration unit				5	80		
estimated HTHP potential 2016			4.0				
DH production with HTHP 2016			0.6				

*0.85 TWh/a per 100 MW installed datacenter capacity.

Table 4. District heating energy potentials: references.

DH network			[150]	[79]	[68, 87]	[67]	[67]
wastewater	[149]			[151]	[153]		
datacenter	[83]			[83]	[83]		
CCHP				[88]	[88]		
Flue gas				[18, 40]	[40, 82, 139]		
refrigeration unit				[152]	[75, 152]		
estimated HTHP potential 2016			[151]				
DH production with HTHP 2016			[151]				

Appendix 3. Demanded temperatures in industrial processes, Temperature = $T_{sink,out}$. [8,17]



Appendix 4. Compressor displacements and pressures

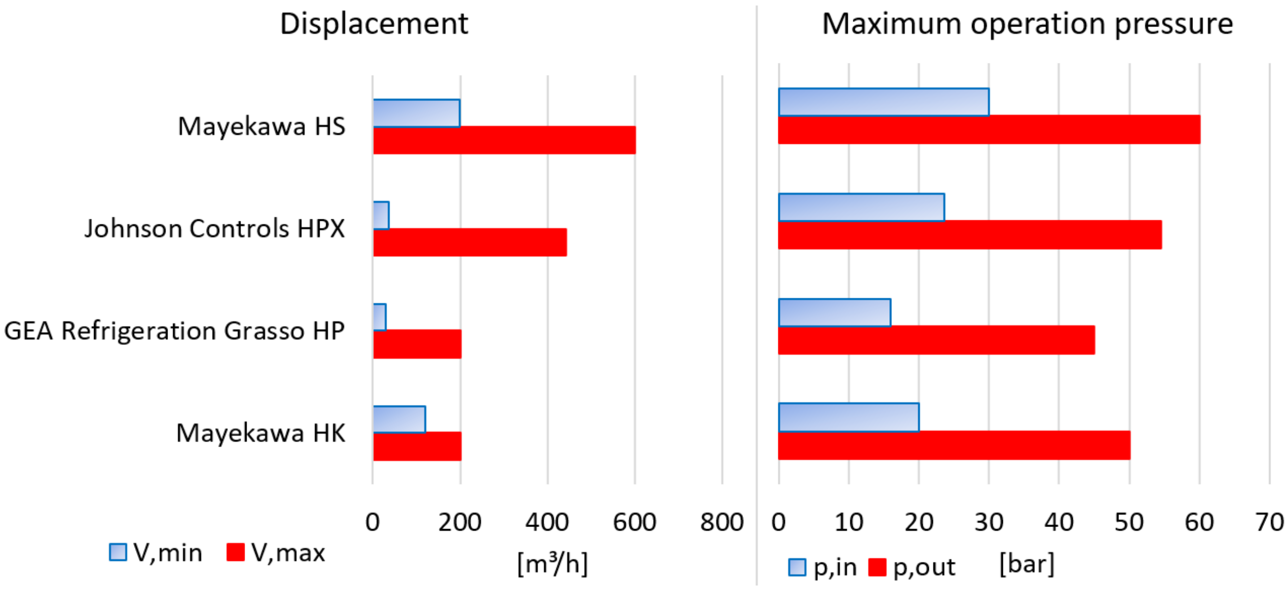


Figure 1. R717 piston compressors.

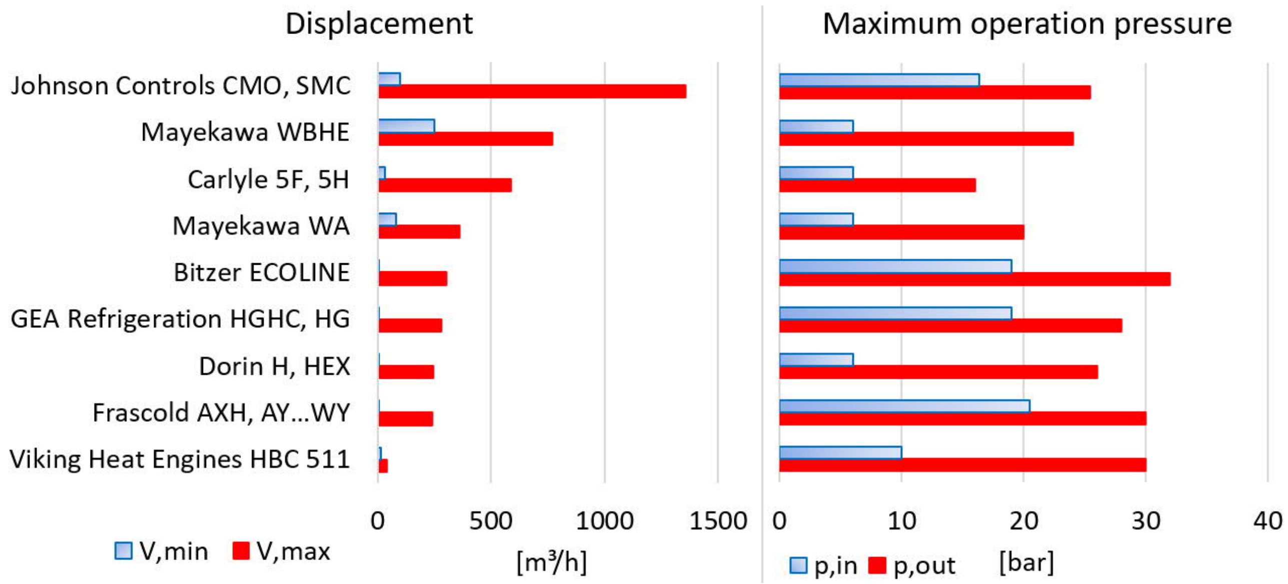


Figure 2. HC/HFO/HCFO piston compressors.

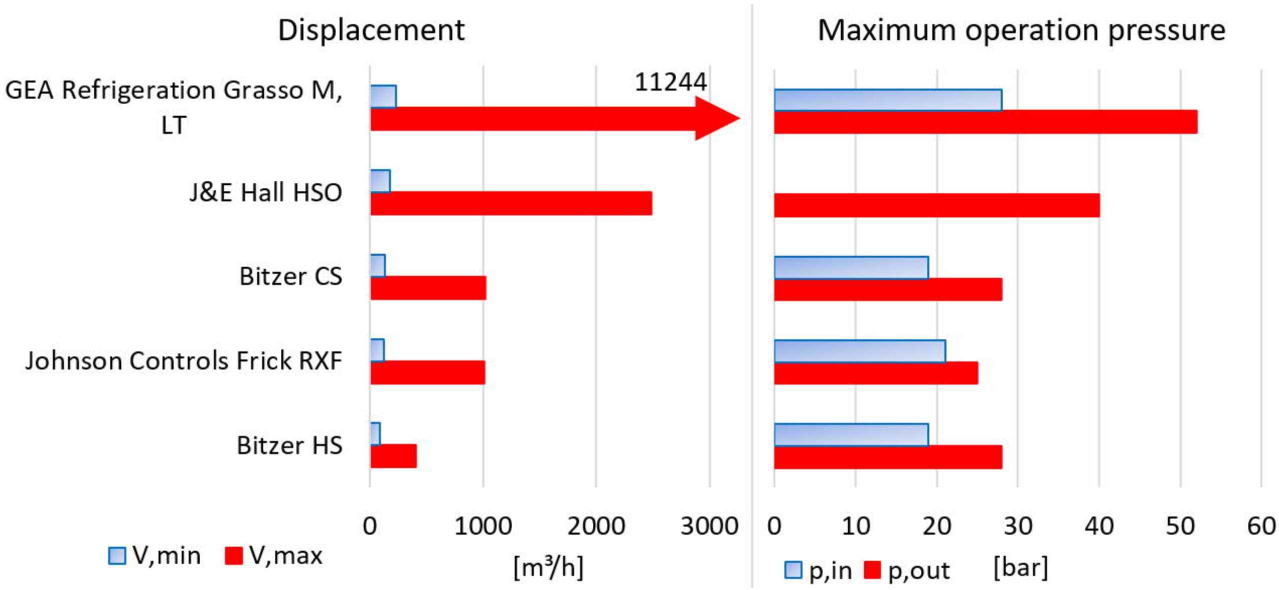


Figure 3. HC/HFO/HCFO screw compressors.